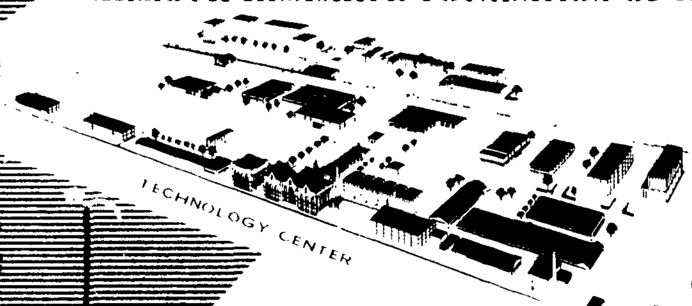
258627

CATALOGED BY ASTIA
AS AD No. ______

ARF

ARMOUR RESEARCH FOUNDATION OF ILLINOIS INSTITUTE OF TECHNOLOGY



Contract No. DA-11-022-ORD-905
OCO, R & D Branch Project No. TB5-2
Department of Army Project No. 593-21-060

DEVELOPMENT OF QUALIFICATION TEST METHODS FOR GEAR LUBRICANTS

Final Summary Report

Conducted and Prepared by
AutoResearch Laboratories, Incorporated

for
Armour Research Foundation

under Project 401

25 years of research

$ 11.50

XOX

Conducted for
Armour Research Foundation

by

AutoResearch Laboratories, Incorporated
Chicago 38, Illinois

DEVELOPMENT OF QUALIFICATION TEST METHODS

FOR GEAR LUBRICANTS

Project 401

Final Summary Report

Contract No.:
DA-11-022-ORD-905

OCO R & D Division Project TB5-2
Dept. of Army Project No. 593-21-060

Submitted by:

D. L. Powell, Project Engineer
AutoResearch Laboratories, Incorporated

This report has been reviewed and approved for the Chief of Ordnance

R. E. Streets

FINAL SUMMARY REPORT

DISTRIBUTION LIST

Copies	Recipient
5	Office, Chief of Ordnance Department of the Army Washington 25, D. C. Attention: ORDTB
2	Commanding Officer Frankford Arsenal Philadelphia 37, Pennsylvania
2	Commanding Officer Watertown Arsenal Watertown 72, Massachusetts Attention: Laboratory
5	Central Air Documents U. B. Building Dayton 2, Ohio Attention: CADO-D
1	Chicago Ordnance District 623 South Wabash Avenue Chicago 5, Illinois Attention: ORDEC-DR
2	Armour Research Foundation Main Files - H. J. Schulthesz
2	AutoResearch Laboratories, Incorporated

ARMOUR RESEARCH FOUNDATION IS AUTHORIZED TO FURNISH COPIES OF THIS REPORT TO ONLY THOSE NAMED ABOVE. OTHER PARTIES OR AGENCIES DESIRING COPIES MUST ADDRESS THEIR REQUESTS TO THE FIRST NAMED ABOVE.

DEVELOPMENT OF QUALIFICATION TEST METHODS

FOR GEAR LUBRICANTS

ABSTRACT

A multi-phase experimental and analytical project has been undertaken to develop test procedures for the evaluation of hypoid gear lubricants and to study the loading, temperature and sliding velocity conditions which exist on automotive hypoid gears. Each of the six phases of this work are discussed in individual sections of this report.

The work included in this project conducted in the period from 1952 to 1960 is as follows:

I. Development of Moisture Corrosion Test Technique

II. Study of Dynamic Loading of Automotive Hypoid Gears

III. Study of Sliding Velocity of Automotive Hypoid Gears

IV. Study of Unit Surface Loading of Hypoid Gears

V. Study of Dynamic Surface Temperatures of Hypoid Gears

VI. Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations.

TABLE OF CONTENTS

	Page
ABSTRACT	iii
I. INTRODUCTION	1
II. PHASE I - DEVELOPMENT OF MOISTURE CORROSION TEST TECHNIQUE	1
A. Object of Program	1
B. Method of Development	2
C. Results and Discussion	5
1. Discrimination	5
2. Repeatability	6
3. Reproducibility	12
4. Study of the Effect of Test Variables	15
III. PHASE II - STUDY OF DYNAMIC LOADING OF AUTOMOTIVE HYPOID GEARS	33
A. Introduction	33
B. Study of Dynamic Loading in Standard Gear Lubricant Tests	34
1. Object	34
2. Method	34
3. Results and Discussion	36
4. Conclusions	46
C. Study of Dynamic Loading in Army Field Test at Yuma, Arizona	48
1. Object	48
2. Method	48
3. Results and Discussion	52
4. Conclusions	66
IV. PHASE III - STUDY OF SLIDING VELOCITY OF HYPOID GEARS	67
A. Introduction	67
B. Object of the Study	68

TABLE OF CONTENTS (Contd.)

		Page
	C. Gears Studied	68
	D. Method of Computation	69
	E. Results and Discussion	70
	1. Sliding Relationships	70
	2. Comparison of Sliding and Load in Full-Scale Lubricant Tests	75
	3. Sliding Velocity of Army Truck Gears in Yuma Field Tests	84
	4. Present Gear Design Trends and Their Effects on Sliding	89
	F. Conclusions	97
	1. General	97
	2. Gear Sliding in Laboratory Evaluation Techniques	97
	3. Gear Sliding in the Yuma Tests	98
V.	PHASE IV - STUDY OF UNIT SURFACE LOADING OF HYPOID GEARS	98
	A. Introduction	98
	B. Methods of Study	99
	C. Results and Discussion	99
	1. Theoretical Study	99
	2. Experimental Determination of Contact Area	103
	D. Future Work Required	104
VI.	PHASE V - STUDY OF DYNAMIC SURFACE TEMPERATURE OF HYPOID GEARS	106
	A. Introduction	106
	B. Future Work	108
VII.	PHASE VI - COOPERATION WITH THE COORDINATING RESEARCH COUNCIL	108
	A. Introduction	108
	B. Discussion of Work Undertaken	109

TABLE OF CONTENTS (Contd.)

	Page
1. CRC L-19 Axle Buildup Procedure	109
2. Development of CRC L-37 Test	110
3. Development of CRC L-42 Test	110
4. Field Surveys and Investigations	111
VIII. ACKNOWLEDGEMENTS	111

APPENDIX A - PROPOSED TEST TECHNIQUE FOR DETERMINATION OF MOISTURE CORROSION CHARACTERISTICS OF AUTOMOTIVE GEAR LUBRICANTS

APPENDIX B - CRC L-19 TEST - TABULATED DATA OF LABORATORY AND FIELD TORQUE RECORDINGS

APPENDIX C - TABULATED DATA OF GEAR LOADINGS IN ARMY FIELD TESTS

APPENDIX D - BLOCK DIAGRAM OF DIGITAL COMPUTER PROGRAM FOR HYPOID GEAR SLIDING COMPUTATION

APPENDIX E - DERIVATION OF EXPRESSION FOR RADIUS OF CONTACT & MAXIMUM PRESSURE BETWEEN A SPHERE & A FLAT PLATE

APPENDIX F - A STANDARD TECHNIQUE FOR REBUILDING CHEVROLET TEST AXLES FOR THE CRC L-19 TEST

LIST OF ILLUSTRATIONS

Figure		Page
1	Cover Plate From One-Day Moisture Corrosion Test on RGO 48-54	7
2	Cover Plate From One-Day Moisture Corrosion Test on RGO 50-54	8
3	Cover Plate From One-Day Moisture Corrosion Test on RGO 52-54	9
4	Cover Plate From One-Day Moisture Corrosion Test on RGO 100-57	10
5	Cover Plate From One-Day Moisture Corrosion Test on RGO 110-57	11
6	Cover Plate From One-Day Moisture Corrosion Test Using Cork Gasket on RGO 110	17
7	Cover Plate From One-Day Moisture Corrosion Test Using Victorite Gasket on RGO 110	18
8	Cover Plate From One-Day Moisture Corrosion Test Using Manila Paper Gasket on RGO 110	19
9	Cover Plate From One-Day Moisture Corrosion Test Using Brown Paper Gasket on RGO 110	20
10	Cover Plate From One-Day Moisture Corrosion Test Using Neoprene Gasket on RGO 110	21
11	Cover Plate From CRC L-33 One-Day Moisture Corrosion Test on RGO 110-57 With Pre-Soaked Gasket - Run 1	25
12	Cover Plate From CRC L-33 One-Day Moisture Corrosion Test on RGO 110-57 With Pre-Soaked Gasket - Run 2	26
13	Strain Gages and Slip Ring Assembly on Automobile Axle Shaft	35
14	Strain Gages on Automobile Torque Tube	35
15	Comparison of Peak Ring Gear Torques in Road and Laboratory L-19 Tests	38
16	Effect of Spark Plug Fouling in Laboratory L-19 Test	40
17	Effect of Throttle Actuation Rate in Laboratory L-19 Test (Throttle Opened Suddenly at 10 MPH)	42
18	Comparison of Peak Ring Gear Torques in L-19 and G.M. Shock Test on Road	44
19	Coast Side Shock Loads at 65 MPH in High Gear Road GM Shock Test	45
20	Coast Side Shock Loads at 45 MPH in Second Gear Road GM Shock Test	47

LIST OF ILLUSTRATIONS (Contd.)

Figure		Page
21	Arrangement of Instrumentation in M-37 Truck	50
22	Effect of Grades on Truck Axle Torques (Yuma Gear Lubricant Test Course)	53
23	Typical Gear Torques in M-37 Truck During Field Tests	56
24	Average Effect of Trailed Load on Axle Torque in Hill Climbing	60
25	Ring Gear Torque in M-211 Truck Due to Gear Shifting (Operating in Low Range)	62
26	Ring Gear Torque in M-211 Truck During Gear Shift From High 2nd to Low 4th	65
27	Direction and Velocity of Sliding on M-37 Truck Ring Gear	71
28	Ridging Failure on M-37 Truck Ring Gear	73
29	Comparison of Angle of Slide Across Profile for Four Automotive Gears	74
30	Comparison of Specific Sliding Velocity for Four Automotive Gears	76
31	Comparison of Unit Gear Loading of CRC L-19 and L-42 High Speed Gear Lubrication Evaluation Procedures	77
32	Comparison of Maximum Gear Sliding Velocity in CRC L-19 and L-42 High Speed Gear Lubrication Evaluation Procedures	78
33	Comparison of Maximum Gear Sliding Velocity and Load in CRC L-20 and L-37 Evaluation Procedures	83
34	Comparison of Sliding Velocity of M-37 and M-211 Truck Gears in Yuma Field Tests	86
35	Pinion Offset of American Passenger Automobile Hypoid Gears	91
36	Trend in Gear Ratio of Passenger Cars	92
37	Effect of Pinion Offset on Sliding Velocity (Hypothetical Gear Set)	94
38	Effect of Gear Ratio on Sliding Velocity (Hypothetical Gear Set)	96
39	Theoretical Line of Contact and Swept Area of Contact for Hypoid Gear	100

LIST OF ILLUSTRATIONS (Contd.)

Figure		Page
40	Theoretical Area of Contact for Hypoid Gears	101
41	Actual Contact Area of Spiral Bevel Gears Under Load	102
42	Maximum Pressure and Radius of Contact of a Half Inch Diameter Steel Ball on a Flat Plate	105

LIST OF TABLES

Table		Page
1	Discrimination of Moisture Corrosion Protection Ability of Several Reference Gear Lubricants	6
2	Repeatability of Moisture Corrosion Test with Several Reference Gear Lubricants	13
3	Results of Cooperative Program on Reproducibility of One-Day Moisture Corrosion Test Procedure	14
4	Results of One-Day Moisture Corrosion Tests With Various Gasket Materials	22
5	Water Vapor Absorption of Gasket Materials	22
6	Study of Effect of Axle Venting on Corrosion of Axle Cover Plates	28
7	Comparison of High Speed Shock Test Gear Conditions	79
8	Comparisons of High Torque Steady Speed Test Gear Conditions	82
9	Average Gear Loads and Sliding Velocities on M-37 Vehicle Yuma Highway Portion Test Course	87
10	Average Gear Loads and Sliding Velocities on M-211 Vehicle Yuma Highway Portion Test Course	87
11	Average Gear Shock Loads and Sliding Velocities in M-37 Truck Due to Gear Shifting	88
12	Average Gear Shock Loads and Sliding Velocities in M-211 Truck Due to Gear Shifting	88

DEVELOPMENT OF QUALIFICATION TEST METHODS

FOR GEAR LUBRICANTS

I. INTRODUCTION

This is the final report of work conducted under Ordnance Contract No. DA-11-022-ORD-905 since the inception of the program in 1952. The work on this program has been conducted in a number of phases as follows:

Phase I	Development of Moisture Corrosion Test Technique
Phase II	Study of Dynamic Loading of Automotive Hypoid Gears
Phase III	Study of Sliding Velocity of Automotive Hypoid Gears
Phase IV	Study of Unit Surface Loading of Hypoid Gears
Phase V	Study of Dynamic Surface Temperatures of Hypoid Gears
Phase VI	Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations

Since the subject matter of each of the phases are somewhat independent of each other, this report treats each of the phases separately.

In the conducting of this program several summary progress reports have been prepared and submitted on certain phases of the program. Reference is made herein to the appropriate summary progress report for detailed data and thorough description of test methods and apparatus used. This report will be confined to a general discussion of the object of each phase, the method of testing, the general results, along with a discussion of the results and the conclusions drawn from the work.

II. PHASE I - DEVELOPMENT OF MOISTURE CORROSION TEST TECHNIQUE

A. Object of Program

During the recent Korean conflict, the Army experienced numerous failures of trucks and other vehicles from the corrosion of rear axle components when the vehicles had been "dead lined" prior to shipment to Japan for periodic maintenance and repair. It appeared that the storage of these vehicles in humid areas, especially close to the sea coast, resulted in corrosion by the natural thermal breathing of the axle housings in the temperature cycling of the atmosphere. It became apparent that the multipurpose gear lubricants being purchased by the government had insufficient ability to protect the axle components from this type of moisture corrosion.

The moisture corrosion test procedure in use at that time for the qualification of rear axle lubricants involved the operation of the candidate lubricant in a Chevrolet axle for four hours at 180 $\pm$ 2°F and then the storage of that axle at room temperatures in the laboratory atmosphere for ten days. The repeatability of this test technique on any one lubricant left much to be desired, probably because of the lack of control over the atmosphere in which the axle was stored but also because of the lack of uniformity in the preparation of the test parts.

This program was initiated with the purpose of providing a test technique for use by the government to differentiate more accurately between abilities of lubricants to protect axles against moisture corrosion.

B. <u>Method of Development</u>

The initial study in this program was to determine whether any small scale bench corrosion test apparatus would show promise of satisfactorily meeting the objectives of this program. Because of the varieties of ferrous alloys found in a rear axle, the complex interaction of the metallic surfaces and gear lubricant components under axle operating conditions, it was quickly decided that in order to produce tests equivalent to results obtained in full-scale trucks and vehicles on the road it would be necessary to use an automotive differential as a test device.

As a second step it was necessary to select some reference lubricants which were known to have varying tendency toward corrosion in rear axles. This selection was complicated by the fact that very sketchy service data were available on the moisture corrosion performance of particular lubricants. After extended experimentation, some reference lubricants were produced containing additives which were known to be particularly severe corrosion producers. As the test development continued and refinements were produced in the test technique and apparatus, a variety of lubricants were obtained which gave a broad spread in resultant corrosion in the eventual test technique.

For some time it had been apparent to a number of investigators that the atmosphere in the then currently used CRC L-21 moisture corrosion test was not adequately controlled to give proper repeatability and reproducibility in that test. It was also apparent that the observed field corrosion had occurred in areas of high relative humidity. It was reasoned that the induction of moisture laden air into the axle was facilitated by the normal temperature-humidity daily cycle. That is, upon cooling in the evening, the air within the axle housing contracted drawing the moist ambient air into the axle housing where it condensed on the colder axle parts during the night. In order to reproduce this type of condition in the proposed test, the axle was installed in a closed box which contained air at 100 per cent relative humidity. Attempts were made to induce the moist air into the axle first by temperature cycling and second by forced aspiration. However, it was noted in both of these cases that the relative humidity of the air inside the axle was depressed below the dew point. No conclusive explanation was found for this phenomenon; however, it was decided that higher humidity air could be retained in the axle and a more uniform relative humidity maintained if the axles were sealed with a small amount of distilled water entrained in the test lubricant. Subsequent experiments with this procedure verified the preliminary assumptions. All future experimentation from that point was conducted with distilled water entrained in the test lubricant.

Due to the variation which was found in production axle assemblies in the preloading of the side carrier bearings and the resultant stabilized running temperature of the test axle, it became apparent that a means was necessary for controlling the temperature of the test axle during motoring at the beginning of the test. In addition, experimentation had shown that greater amounts of corrosion could be produced if the axle was first run at an elevated temperature and then cooled to a lower storage temperature for a period of time. It was also apparent that this was the field operating cycle under which corrosion had been produced. For these reasons a temperature cycle was developed which called for motoring operation of the axle at 180°F followed by a quiet storage period at 125°F. With this cycling procedure, it is characteristically true that the cover plate of the axle has liquid phase moisture covering it when the axle is disassembled at the end of the storage period as long as ten days later.

Upon establishment of the proposed test cycle, a series of studies were initiated to determine the effect of variations in the test operating conditions and to determine the repeatability of the test technique with a selected reference oil. In addition, the ability of the test to discriminate between oils was demonstrated by a series of tests involving a wide variety of axle lubricants. Corrosion on the axle cover plate ranged from completely clean to 100 per cent covered with a heavy orange and black corrosion coat depending upon the reference lubricant used.

A detailed outline of the final test technique and a description of the test apparatus is presented in Appendix A.

C. Results and Discussion

1. Discrimination

Of prime importance in the development of this test procedure was the establishment of conditions and techniques which would provide clear discrimination between the lubricants with varying ability to protect the axle against moisture corrosion. On the basis of some field test experience, three CRC reference gear lubricants were selected as lubricants one of which would be expected to give very good moisture corrosion protection and the other two poor moisture corrosion protection. These lubricants were: RGO 48-54, RGO 50-54 and RGO 52-54. In addition to these lubricants, a wide variety of other MIL-L-2105 quality level lubricants and various factory fill lubricants were tested in the proposed procedure. Large differences in test results were observed with these various lubricants. Table 1 summarizes the results on the three primary reference oils used in the study along with the results from CRC reference oils RGO 100-57 and 110-57. It may be observed that the test technique differentiates between these lubricants both as to extend and degree of corrosion. Figures 1 through 5 are photographs of representative cover plates from the tests reported in Table 1. From these photographs it may be seen that visual rating of the differentiation between these lubricants is not difficult.

Table 1

DISCRIMINATION OF MOISTURE CORROSION PROTECTION ABILITY OF SEVERAL REFERENCE GEAR LUBRICANTS

Reference Oil	Test Duration, Days	Avg. Area Corroded,%	No. of Tests	Nature of Corrosion
RGO 48-54	1	0	4	-
RGO 48-54	10	0	1	-
RGO 50-54	1	75	4	moderate to heavy orange and black
RGO 52-54	1	62	9	light to moderate black
RGO 100-57	1	66	4	widely scattered light orange and black corrosion
RGO 100-57	7	63	3	light to heavy scattered orange and black corrosion
RGO 110-57	1	Trace	4	scattered spots at gasket edge
RGO 110-57	7	3	4	brown and black spots in upper portion and along gasket edge

2. Repeatability

The repeatability of the test technique was investigated by replicate tests using reference oils RGO 52-54, RGO 100-57 and RGO 110-57. Table 2 gives the results of these tests. It may be seen that the spread in results with reference oil RGO 100-57 was from 50 per cent to 80 per cent surface covered. One-day tests with reference oil RGO 110-57 consistently produced a trace of corrosion on the cover plates. In the seven-day test results ranged from 2 to 5 per cent covered. The results of the tests conducted on RGO 52-54 ranged between 50 and 70 per cent of the cover plate covered by corrosion.

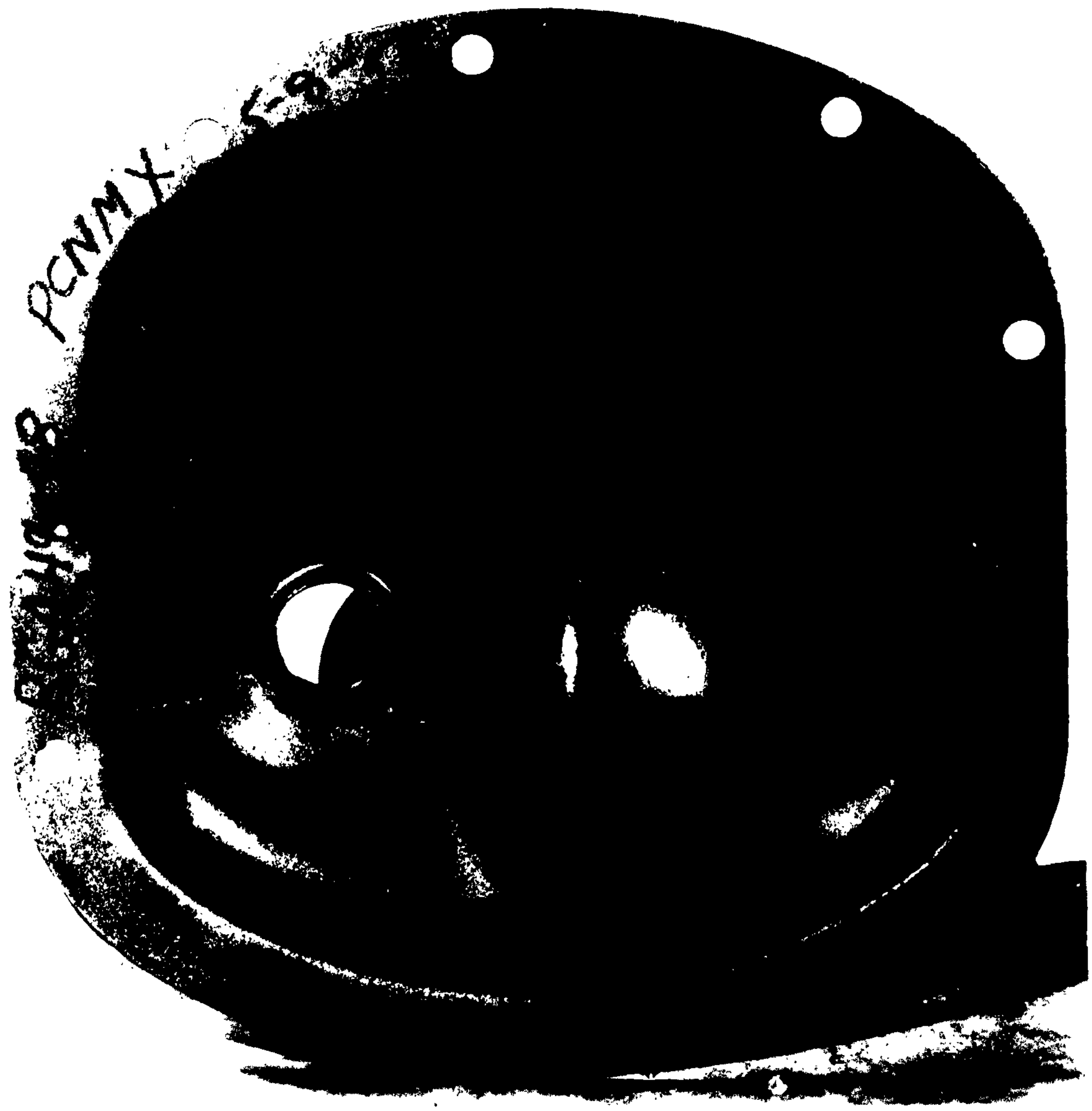
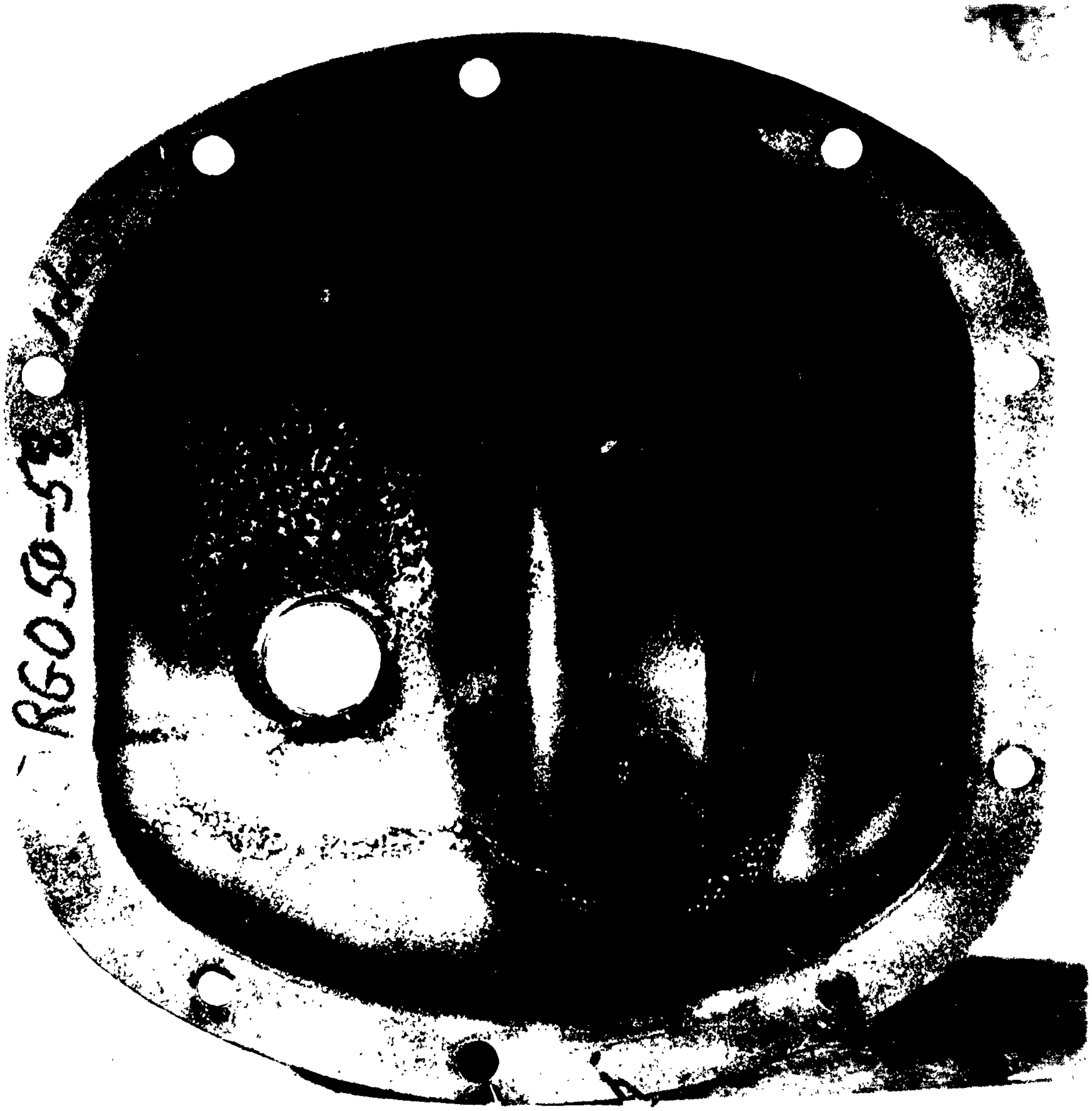


Fig. 1 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST ON RGO 48-54



<u>Fig. 2</u> <u>COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST ON RGO 50-54</u>

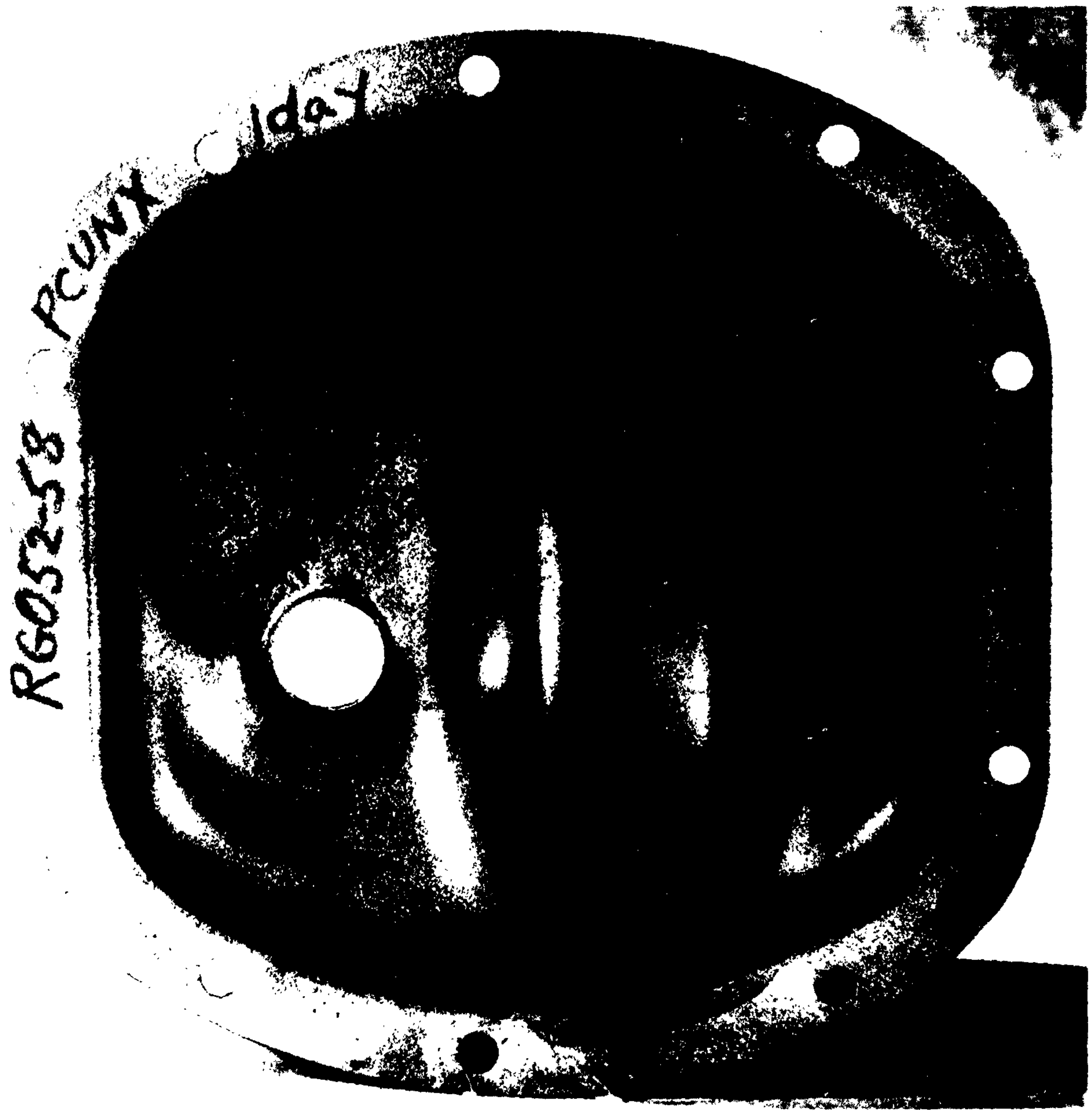


Fig. 3 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST ON RGO 52-54

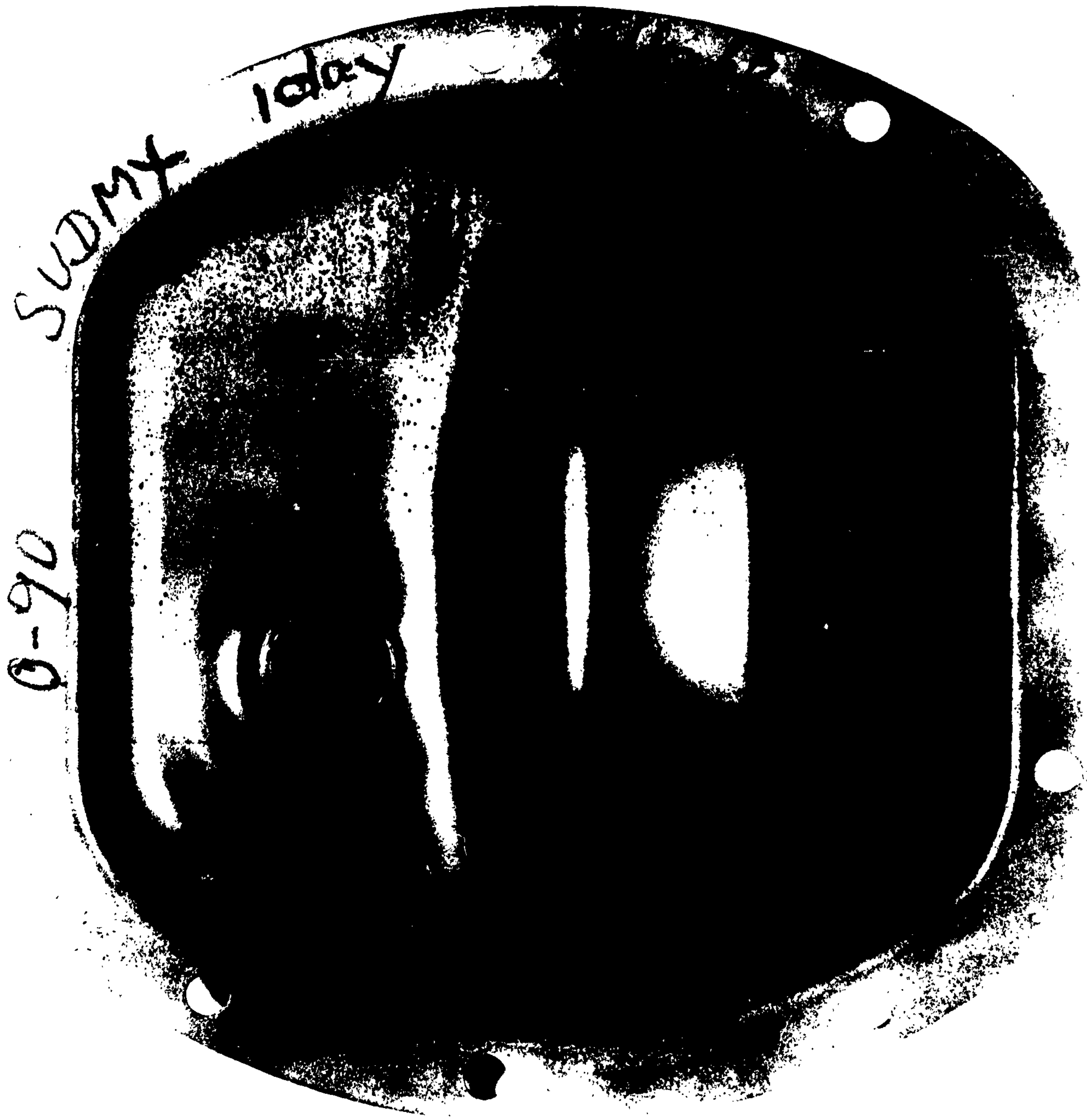


Fig. 4 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST ON RGO 100-57

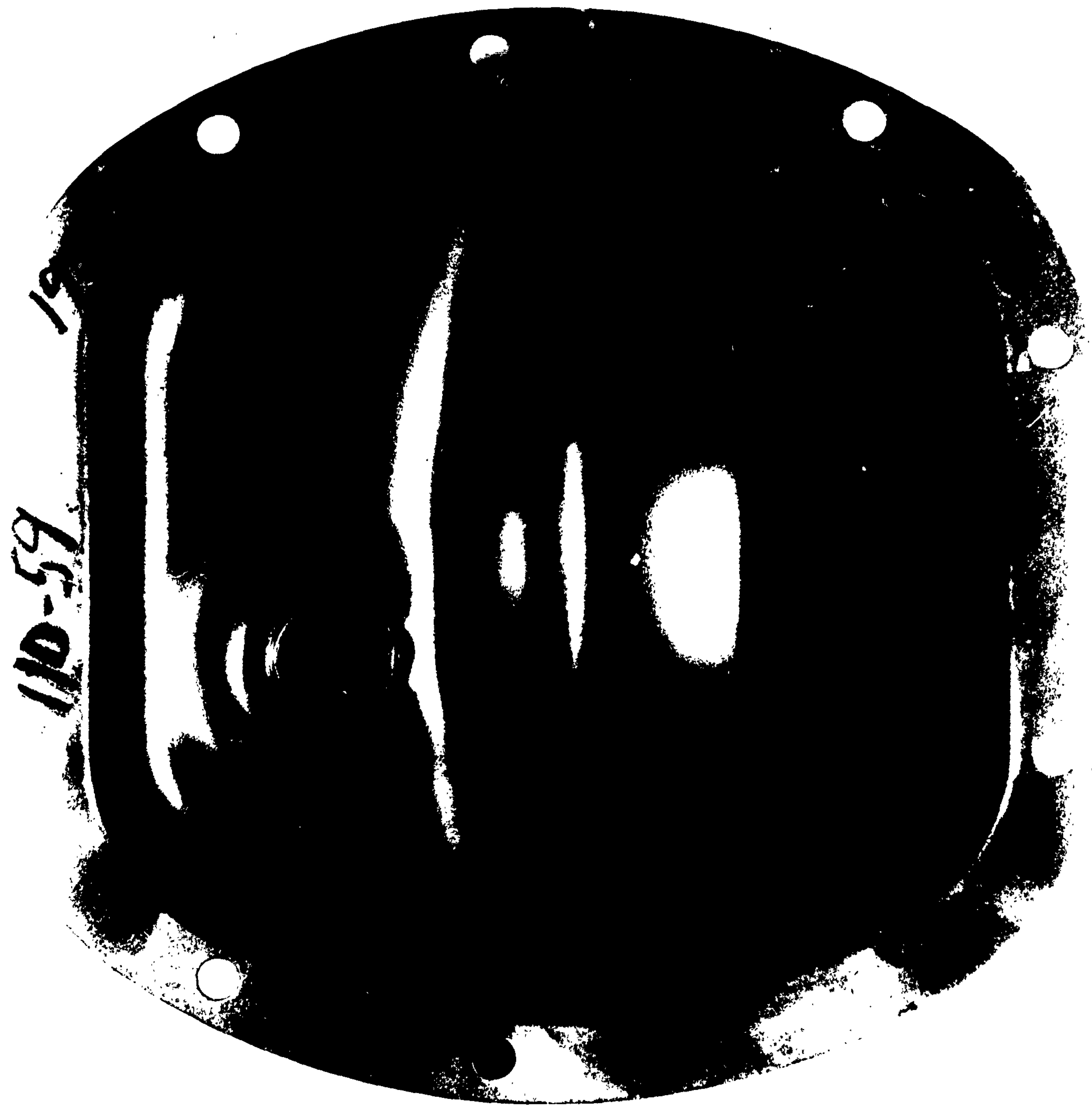


Fig. 5 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST ON RGO 110-57

Whereas with the reference lubricants which produced sizable amounts of cover plate corrosion, the spread in results which were obtained was as much as 30 per cent, it will be noted that with lubricants which produce small amount of corrosion the spread in corrosion is proportionately reduced. Since the acceptable limit for area corroded on the axle cover plate is in the range of that produced by lubricant RGO 110-57, it would appear that the repeatability of the proposed procedure is adequate to successfully test the lubricants for qualification under military specifications.

3. Reproducibility

In conjunction with the CRC Gear Lubricants Group, a program was run to determine the reproducibility of the moisture corrosion test as run in four laboratories. The five primary reference oils used in the study on discrimination were also used in this program. Since it had been observed that, in general, the results of the seven-day test differed from the one-day test only in the degree of corrosion formed, the one-day test was used as the tool to determine the reproducibility of the procedure.

The results of the reproducibility program are presented in Table 3. It may be noted that with the exception of one laboratory, the results obtained in the four cooperating laboratories on all of the reference oils appear to fall within the range of repeatability as established at the Armour Research Foundation. Subsequent consultation with representatives of that laboratory revealed that there were certain differences in the apparatus and technique which they used in conducting this test. Upon revision of their apparatus and technique to conform to the procedure outlined in Appendix A, results of reference oil tests

Table 2

REPEATABILITY OF MOISTURE CORROSION TEST WITH SEVERAL REFERENCE GEAR LUBRICANTS

Reference Oil	Test Duration, Days	Area Corroded, %	Nature of Corrosion
RGO 52-54	1	65	light to moderate black
RGO 52-54	1	75	light to moderate black
RGO 52-54	1	60	light to moderate black
RGO 52-54	1	75	light to moderate black
RGO 52-54	1	50	light to moderate black
RGO 52-54	1	60	light to moderate black
RGO 52-54	1	40	moderate black
RGO 52-54	1	65	moderate black
RGO 52-54	1	65	moderate black
RGO 100-57	1	55	widely scattered orange and black
RGO 100-57	1	50	widely scattered orange and black
RGO 100-57	1	80	widely scattered orange and black
RGO 100-57	1	80	widely scattered orange and black
RGO 100-57	7	65	light to heavy orange and black
RGO 100-57	7	75	moderate to heavy orange and black
RGO 100-57	7	50	widely scattered orange and black
RGO 110-57	1	Trace	scattered spots along gasket edge
RGO 110-57	1	Trace	scattered spots along gasket edge and on upper part of plate
RGO 110-57	1	Trace	scattered spots along gasket edge
RGO 110-57	1	Trace	scattered spots along gasket edge
RGO 110-57	7	2	heavy brown line along gasket edge
RGO 110-57	7	1-2 *	black spots in upper portion
RGO 110-57	7	5	brown spots in upper portion
RGO 110-57	7	4	brown and black spots in upper portion

* A rating of 1-2 per cent of area corroded is considered slightly more than a trace.

conducted in that laboratory fell in line with those obtained at the other three laboratories.

Table 3

RESULTS OF COOPERATIVE PROGRAM ON REPRODUCIBILITY OF

ONE-DAY MOISTURE CORROSION TEST PROCEDURE

Laboratory	Reference Oil	Area Corroded, %	Nature of Corrosion
A	RGO 48-58	None	-
	RGO 48-58	None	-
B	RGO 48-58	None	-
	RGO 48-58	None	-
C	RGO 48-58	None	-
Armour	RGO 48-58	None	-
	RGO 48-58	None	-
A	RGO 50-58	45	heavy orange and black
	RGO 50-58	85	heavy orange and black
B	RGO 50-58	85	heavy orange and black
	RGO 50-58	85	heavy orange and black
C	RGO 50-58	75	heavy orange and black
Armour	RGO 50-58	60	heavy orange and black
	RGO 50-58	80	heavy orange and black
A	RGO 52-58	75	heavy orange and black
	RGO 52-58	75	heavy orange and black
B	RGO 52-58	65	medium to heavy orange and black
	RGO 52-58	50	medium to heavy orange and black
C	RGO 52-58		
Armour	RGO 52-58	40	medium to heavy orange and black
	RGO 52-58	65	medium to heavy orange and black
	RGO 52-58	65	medium to heavy orange and black
A	RGO 100-57	60	medium orange and black
	RGO 100-57	75	medium orange and black
C	RGO 100-57	50	widely scattered orange and black
Armour	RGO 100-57	80	widely scattered orange and black
	RGO 100-57	80	widely scattered orange and black
	RGO 100-57	50	widely scattered orange and black
A	RGO 110-57	15	scattered medium orange and black
	RGO 110-57	10	scattered medium orange and black
C	RGO 110-57	Trace	few scattered spots
Armour	RGO 110-57	Trace	few scattered spots
	RGO 110-57	Trace	few scattered spots
	RGO 110-57	Trace	few scattered spots

4. Study of the Effect of Test Variables

A series of investigations were undertaken to establish the effect of a number of variables in test apparatus and technique upon the differentiation and repeatability of the proposed procedure. These investigations were in general in two areas: (a) variations in test apparatus, and (2) variations in test technique or procedure. Each of the items investigated is discussed individually below.

a. Apparatus

(1) Bearing Preload

It is known that the amount of preload on the side carrier bearings of automotive differentials affects the operating temperature of the differential assembly. Since in the moisture corrosion test the test unit is a production model axle with normal variations in axle preloading, it was believed necessary to determine whether the range of axle preloading which could be expected from these units would cause any sizable variation in test results. Two one-day tests were conducted using reference lubricant RGO 52-54. One of the test axles was assembled with an initial differential preload of 15 in-lb to break and 10 in-lb to turn. The other axle unit was assembled with an initial differential preload of 8 in-lb to break and 6 in-lb to turn. In all other respects the two tests were identical. The results of the two tests with varying differential preload were essentially identical. The cover plates were rated as being 60 to 70 per cent covered with light closely spaced black corrosion.

From these tests it was concluded that within the limits of bearing preload of from 8 to 15 in-lb to break and 6 to 10 in-lb to turn there would be no significant difference in the corrosion results from the tests.

(2) Gasket Material

With possessing lubricants improved moisture corrosion protection presented for qualification under MIL-L-002105A (Ord), little or no corrosion occurs on the cover plates. Characteristically the corrosion which does occur is along the upper edge of the cover plate at the meeting line with the gasket. Concern was felt as to whether the material of the gasket would have an effect upon the amount of corrosion formed in that area. It was reasoned that different gasket materials would differ in ph, absorptivity to water vapor and retention of liquid-phase water on the surface of the gasket.

In order to determine the effect of gasket materials on the gasket line corrosion of the moisture corrosion test, several materials were selected and gaskets prepared for testing under this technique. One-day moisture corrosion tests were conducted using each of five gasket materials. The reference oil used in these tests was RGO 110-58. It was known to produce corrosion only along the gasket line in a one-day test. The rating of the variation of corrosion from these tests was on the basis of the length of the line of corrosion produced along the gasket parting line above the oil level. The results of these tests are presented in Table 4. It may be noted that the periphery of corrosion from these tests ranged from 10 to 100 per cent of the cover plate above the oil level. The gasket materials used in these tests in order of decreasing gasket line corrosion were: cork, Victorite (specially treated paper), manila paper (Spicer standard spare part gasket), commercial brown paper and neoprene. The corrosion with the latter two materials was substantially less than that with the first three materials. Photographs of the cover plates from these tests are shown in Fig. 6 through 10.

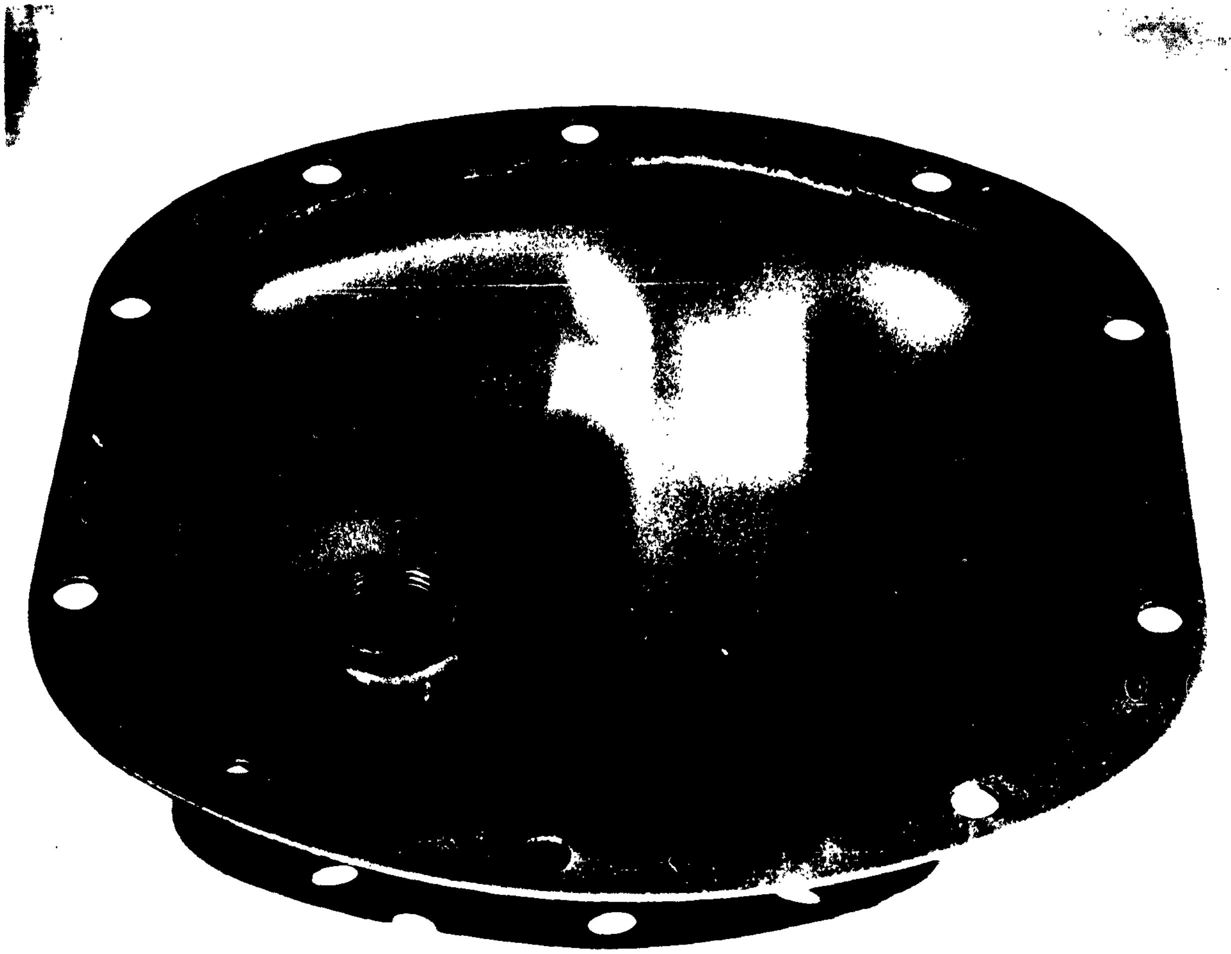


Fig. 6 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST USING CORK GASKET ON RGO 110

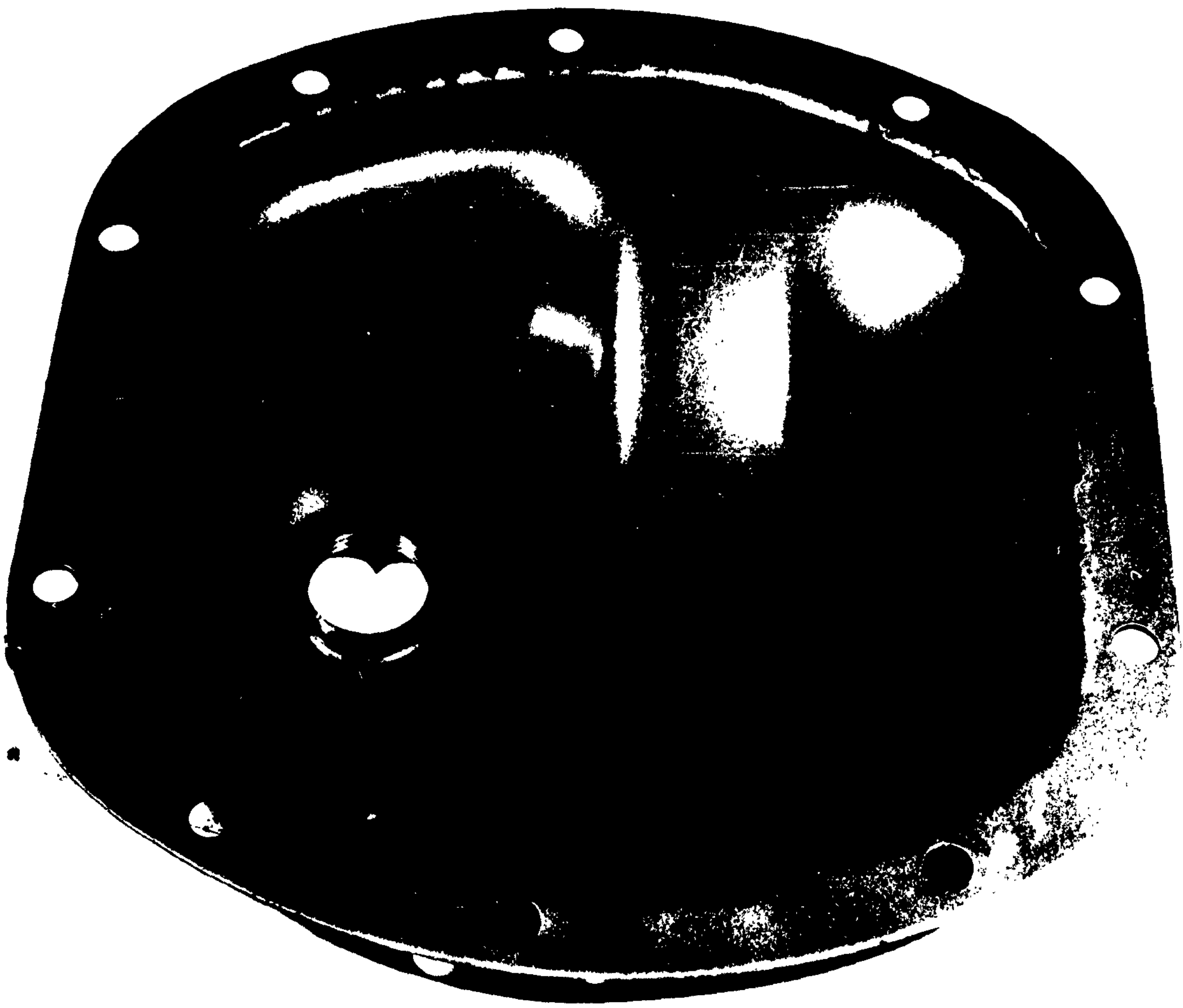


Fig. 7 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST USING VICTORITE GASKET ON RGO 110

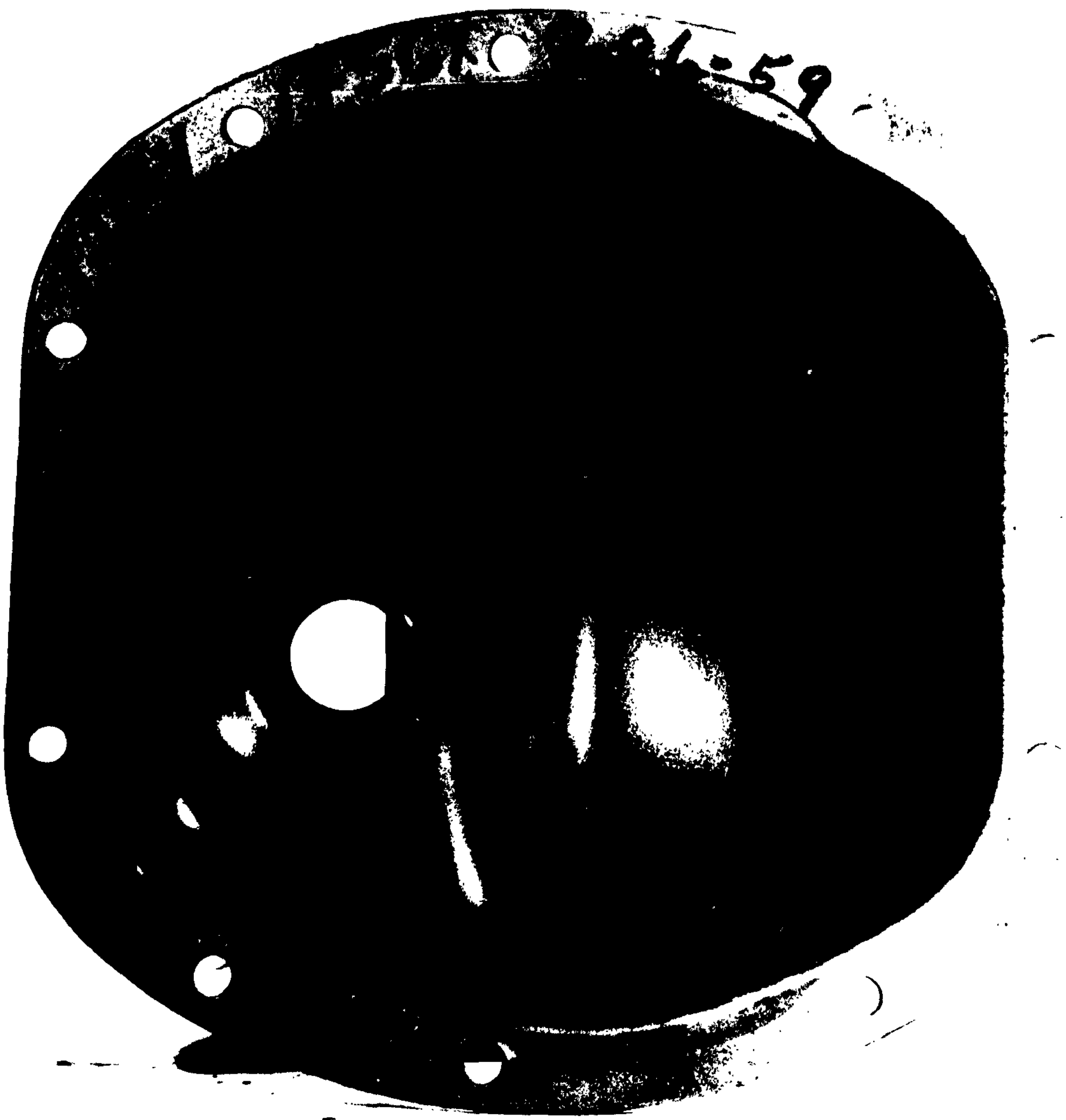


Fig. 8 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST USING MANILA PAPER GASKET ON RGO 110

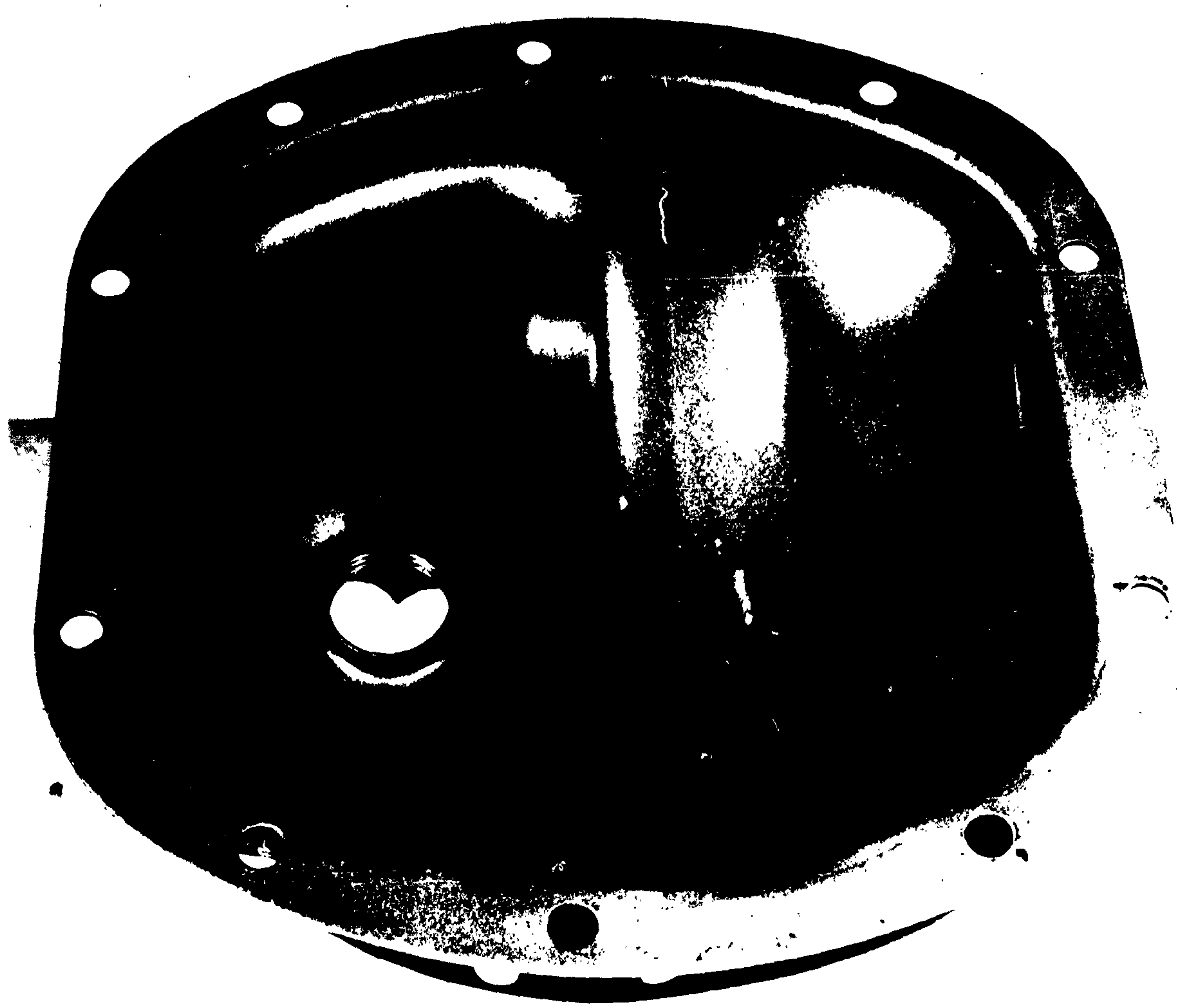


Fig. 9 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST USING BROWN PAPER GASKET ON RGO 110

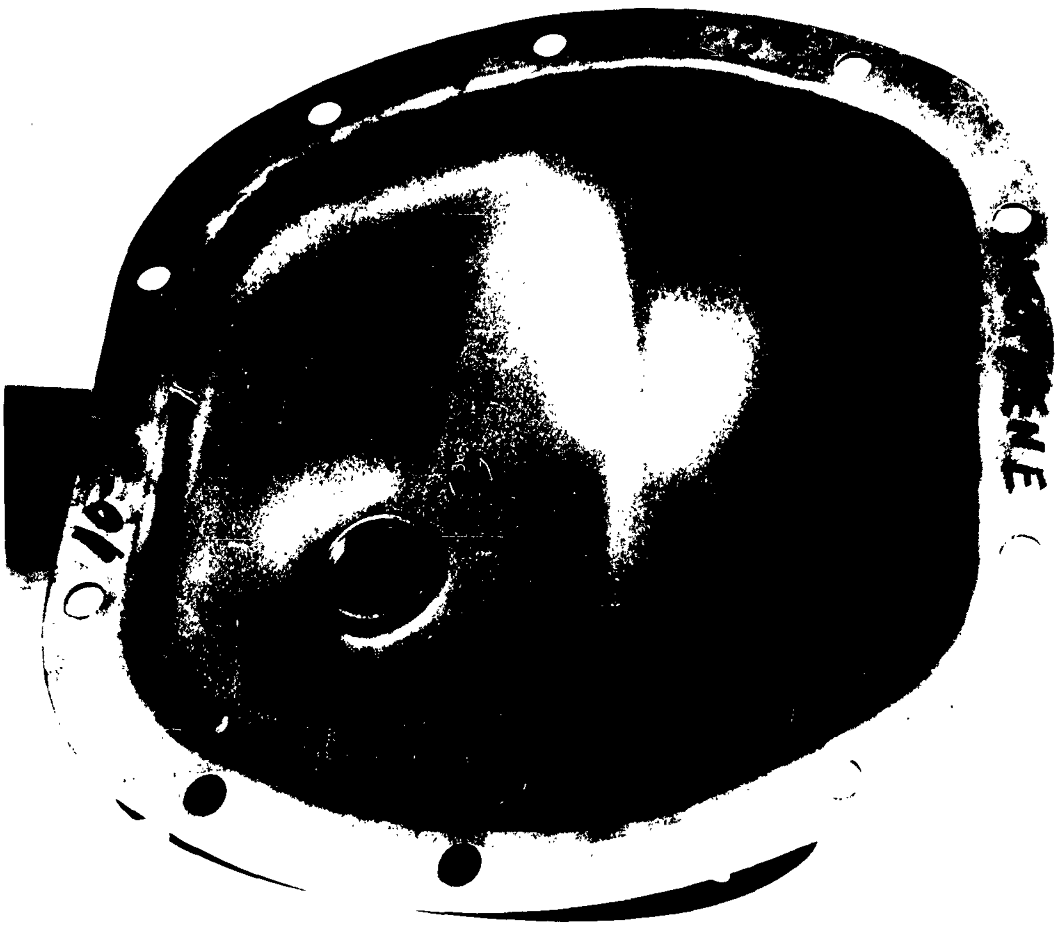


Fig. 10 COVER PLATE FROM ONE-DAY MOISTURE CORROSION TEST USING NEOPRENE GASKET ON RGO 110

Table 4

RESULTS OF ONE-DAY MOISTURE CORROSION TESTS WITH VARIOUS GASKET MATERIALS

Material	Thickness, in.	Corrosion	
		Character	Periphery Corroded Per Cent
Cork	0.127	Medium	100
Victorite	0.023	Light to Medium	75
Manila Paper	0.029	Light	65
Brown Paper	0.004	Light	30
Neoprene	0.030	Light	10

In order to determine the relative tendency of the test materials to absorb water vapor, samples of equal size of each of the gaskets were stored in a bell jar with a desiccant at room temperature for approximately 120 hours to stabilize their moisture content. They were then placed in an atmosphere saturated with moisture for two hours. The samples were weighed before and after exposure to the 100 per cent relative humidity and the weight gain was recorded. The results of these tests are presented in Table 5.

Table 5

WATER VAPOR ABSORPTION OF GASKET MATERIALS

Material	Dry, gms	Weight after 2 hrs. at 100% RH	Weight Increase, gms	Weight Increase Per Cent
Cork	1.1271	1.2162	0.0891	7.8
Victorite	0.6578	0.7918	0.1340	20.3
Manila Paper	0.7186	0.7903	0.0717	10.0
Brown Paper	0.0892	0.0935	0.0043	4.8
Neoprene	1.3347	1.3402	0.0055	0.04

From Tables 4 and 5 it may be observed that with the exception of the cork gasket the greater the per cent increase in weight due to water absorption the greater the amount of peripheral corrosion on the cover plates. It would, therefore, appear that the vapor absorptivity of the gasket material has a direct relationship to the corrosion formed along the gasket edge. However, it would also be assumed that either or both the acidity of the gasket material and its surface porosity would have an important effect on gasket line corrosion.

Since the gasket material can have an appreciable effect upon the results of the moisture corrosion test, it appears important to standardize upon one gasket material for this test work. Since the materials furnished by the manufacturer vary depending upon whether it is a complete differential assembly or whether the gaskets are purchased as a spare part, it has been recommended that the manila paper (Victor No. 27690) gasket which is furnished as a spare part for this unit be established as the standard gasket to be used in all moisture corrosion tests. This, of course, will require that the gaskets supplied in new differential assemblies (Victorite) must be replaced with the manila paper gasket prior to conducting a test.

(3) <u>Gasket Pre-Soaking</u>

Considerable concern has been expressed recently over the exclusive formation of rust along the gasket edge in tests conducted on oils with good moisture corrosion protection. It was questioned whether the gaskets selectively absorbed the moisture out of the oil or from the air and retained it against the surface of the cover plate during the test producing the characteristic line of corrosion along the gasket parting edge. Two one-day tests were conducted to determine whether pre-soaking the gasket in the test oil would prevent the formation of this gasket line corrosion. If this were true, it would then be indicated that the corrosion along the gasket edge was not a fair indication of the moisture corrosion protection afforded by the oil. Photographs of the two test cover plates are shown as Figs. 11 and 12. It may be observed that the results from these duplicate tests with the gasket pre-soaked in test oil are nearly identical to previously run tests on the same reference oil (RGO 110-57) using Victor gasket No. 27690.

As stated earlier, tests have been conducted on several lubricants which produced no corrosion on any part of the cover plate. This was particularly noticed in a large number of tests conducted on RGO 48-54. It would therefore appear that a lubricant allowing a line of corrosion along the gasket edge is measurably inferior in moisture corrosion protection to those lubricants which prevent all corrosion on the cover plate.

In light of the noticeable effect on test results of the absorption of water vapor into the gasket material, the soaking of the test gasket in a pan of test lubricant for a period of approximately five minutes immediately prior to final assembly of the test unit has been established as a standard pretest condition. It is believed that this soaking will help to negate the effect of absorbed moisture in the gasket.

(4) Storage Container Air Flow

During the storage period of the moisture corrosion test the temperature of the test lubricant is controlled by suspending the test axle in a double walled enclosure in which air at a controlled temperature is circulated. Due to the double wall construction of the storage chamber, it was possible to circulate the air either upward or downward over the test

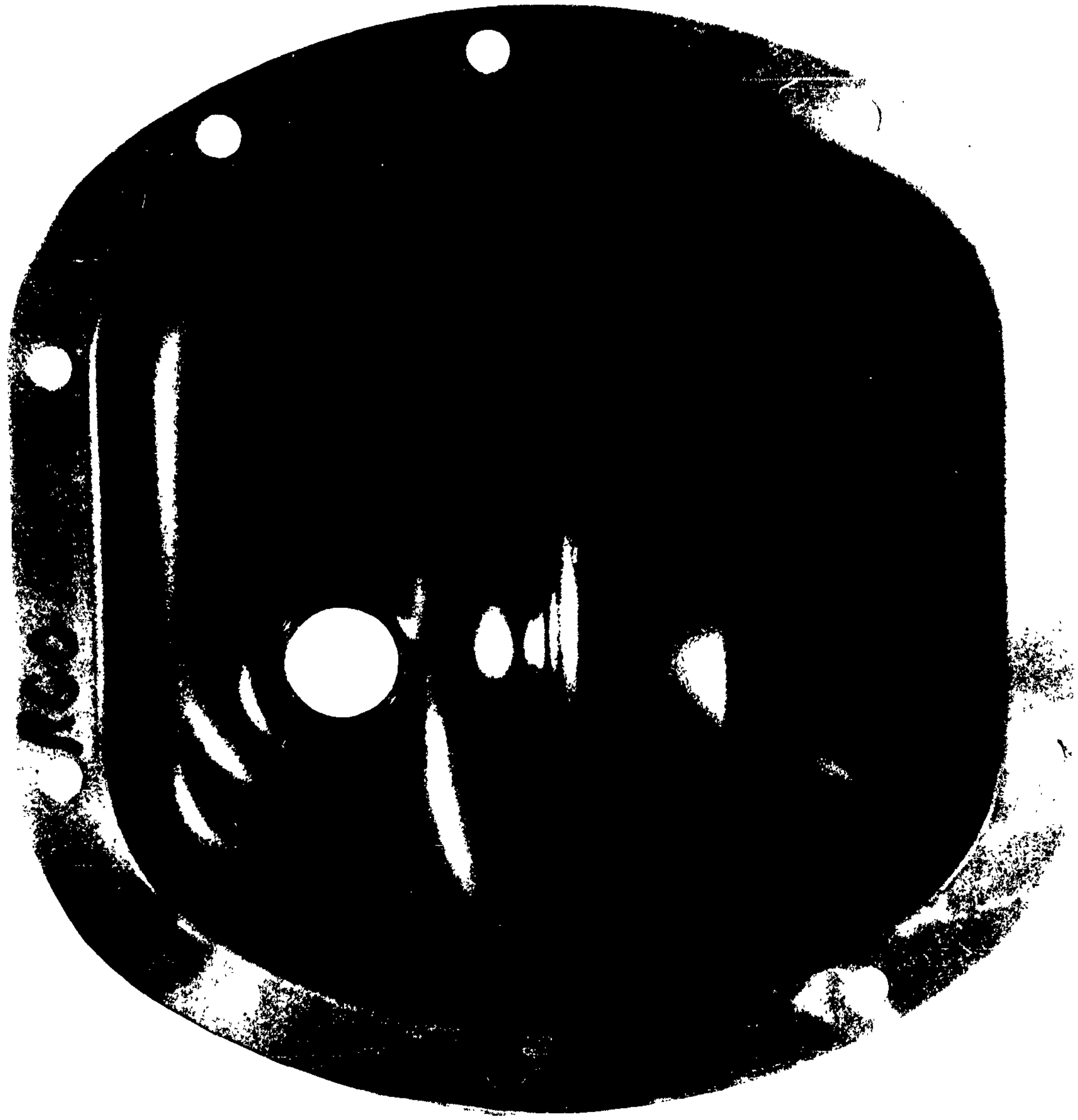


Fig. 11 COVER PLATE FROM CRC L-33 ONE-DAY MOISTURE CORROSION TEST ON RGO 110-57 WITH PRE-SOAKED GASKET - RUN 1

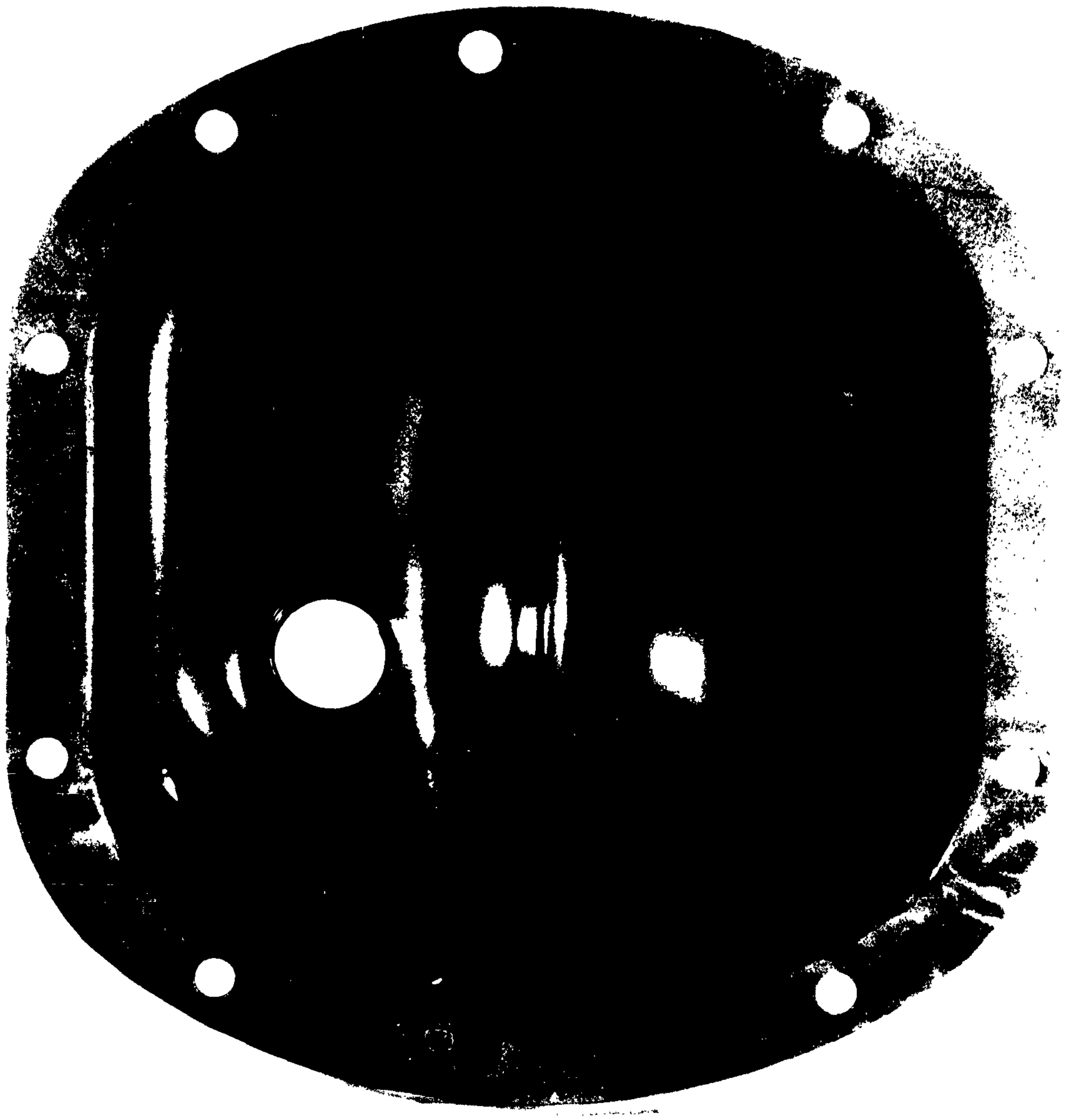


Fig. 12 COVER PLATE FROM CRC L-33 ONE-DAY MOISTURE CORROSION TEST ON RGO 110-57 WITH PRE-SOAKED GASKET - RUN 2

differential by reversing the direction of the fan or by changing the type of fan blades.

A group of four tests were run on two lubricants to determine the effect of these two methods of air circulation. The results of these tests indicate that a more uniform layer of condensation is produced by passing the air up over the test unit from below. When the air is blown down over the test unit the condensation occurs principally at the upper edge of the cover plate and the corrosion produced is concentrated in that area.

As a result of these tests the air flow direction was established in order to provide flow upward over the test unit. A fan impeller of the centrifugal design is used to prevent the possibility of air circulation in the wrong direction.

(5) Venting of Differential

As mentioned under "METHOD OF DEVELOPMENT" early experimentation on a technique had shown that the most repeatable and highest degree of differentiation was obtained in sealing the axle with a small amount of moisture added to the lubricant charge. However, under further investigation it was decided that a study should be conducted to determine the effect of various methods of venting the axle assembly during motoring and the storage phases of the procedure. Table 6 presents the test results from this study. It will be observed that venting of the axle housing has a considerable effect upon the moisture corrosion formed on the axle cover plate. In general, the greater the venting of the axle housing, the less corrosion is formed on the cover plates. The least corrosion was formed when the axle was vented through the 7/64 in. diameter vent in during both motoring and storage phases. The greatest amount of corrosion was formed on the cover plate when the axle was vented through a 5 psig relief valve during the

Table 6

STUDY OF EFFECT OF AXLE VENTING ON CORROSION OF AXLE COVER PLATES

Test Variable					Proposed Procedure
Motoring	Hermetically Sealed		Vented thru 7/64" Dia. Vent to Room Atmosphere		Sealed
Storage	Vented thru 7/64" Dia. Vent to 100% R. H. Atmosphere		Vented thru 7/64" Dia. Vent to 100% R. H. Atmosphere		Sealed
	Run 1	Run 2	Run 1	Run 2	
Duration	1 day	1 day	1 day	1 day	1 day
Lubricant	D *	D	D	D	D
Differential	24	24	24	24	24
Water Addition (cc)	28	28	28	28	28
Motoring Speed (rpm)	2500	2500	2500	2500	2500
Stabilized Motoring, HP	1.16	1.08	1.14	1.12	1.23
Motoring Bulk Oil Temp., °F	177-180	177-180	177-180	177-180	178-181
Storage Temperature, °F	75-80	75-80	75-80	75-80	122
Storage Humidity	100	100	100	100	**
Cover Plate Condition	Light layer of closely spaced orange and black corrosion particles over 40% of the area above the oil level on both tests.		Light layer of closely spaced orange corrosion particles over 30-40% of the area above the oil level on both tests.		Light layer of closely spaced corrosion particles over 50% of the area above the oil level.

* RGO 52-54
** Humidity in the air space above the oil approaches saturation since the moisture cannot migrate from the sealed differential after motoring.

motoring phase and sealed during storage. The increased corrosion observed for the more closely sealed axle assemblies no doubt resulted from the retention of a greater portion of the vapor in the axle housing during the motoring phase and the subsequent corroding of the cover plate by this condensed moisture in the storage phase.

The variation between the corrosion obtained with the pressure relief valve set at 5 psig and 1 psig was not great. Since it was believed that the lower venting pressure more nearly duplicated the condition which exists in automotive equipment in the field, it was decided that the technique should standardize upon the setting of 1 psig relief pressure during the motoring phase. During the storage phase the pressure relief valve is sealed to prevent any migration of the moisture which might occur through a leaking relief valve.

b. Procedural

(1) Motoring Speed

The speed of motoring established for the first portion of the test was 2500 rpm. Realizing in laboratories where this test would be run power requirements fluctuate widely affecting the voltage applied to operating machinery, it was believed necessary to run some reference tests to determine the effect of small changes in speed. Assuming that the operating speed could be maintained within $\pm$ 50 rpm of the nominal 2500 rpm operating speed, two one-day moisture corrosion tests were run with lubricant RGO 52-54 at 2550 rpm and 2450 rpm, respectively. In all other aspects the two tests were identical. These tests both resulted in the formation of a light layer of closely spaced black corrosion over approximately 50 per cent of the area above the oil level. The variation in results between the two tests were negligible. It was, therefore, assumed that variations in speed of $\pm$ 50 rpm would have no measurable effect upon the results of the tests.

(2) Motoring Temperature

The procedure established for this test calls for the temperature of the lubricant in the axle to be maintained at 180 $\pm$ 2°F. In order to determine the sensitivity to variations in motoring temperature, two one-day tests were conducted at 185°F motoring temperature and 175°F motoring temperature, respectively. During the first test the temperature actually varied between 183° and 186°. During the second test the temperature varied between 172° and 175°. The results of these two tests were nearly identical; a light layer of closely spaced black corrosion over approximately 50 per cent of the area above the oil level. From the results of these tests it was concluded that if a test is conducted within the prescribed limits of 180° $\pm$ 2°F no measurable difference will occur in the corrosion formed on the cover plate.

(3) Storage Temperature

The temperature of the oil sump during storage is specified as 125 $\pm$ 2°F. If this temperature is close to the dew point of the air-moisture mixture in the axle a small change in temperature could cause a very large change in the amount of moisture condensed on the cover plate. It would therefore be possible that storage temperature would have a large effect upon the corrosion formed on the cover plates. In order to investigate this effect two one-day moisture corrosion tests were run using reference lubricant RGO 52-54. The tests were run in accordance with the standard procedure except for the storage temperature. In one test the storage temperature was maintained at 130°F; in the other test the storage temperature was maintained at 121°F. The corrosion formed on the cover plate from the test operated at 130°F covered approximately 70 per cent of the area above the oil level. The corrosion formed on the cover plate from the test operated at 121°F covered approximately 50 per cent of the cover plate area above the oil level. From the work conducted on the repeatability of this test with this reference oil, it may be seen that these results are within the limits of repeatability. However, they are at the extreme ends of that range. Therefore, from these two test results it is not possible to conclusively determine whether a temperature change of 9° in storage temperature has a significant effect upon the corrosion produced on the cover plate. However, there is indication that some effect may exist. Therefore, it was considered important to specify in the technique that the storage temperature should be maintained at 125° ± 2°F.

(4) Storage Time

During the development of the test technique several lengths of storage time were used. The principal storage periods were nominally one day, seven days and ten days in length. A one-day test actually involves four hours of motoring and eighteen hours of storage time. The remaining time in the twenty-four hour period is required for initiation of the test. The storage time on a seven-day test is actually six days and eighteen hours. Likewise for a ten-day test the storage time is nine days and eighteen hours.

Experience with the test technique had shown that oils with varying moisture corrosion protection were rated in the same order by a one-day, a seven-day or a ten-day test. However, with the longer storage periods it was possible to develop the existing corrosion somewhat more so that the intensity of the corrosion was greater although the extent of the coverage was essentially the same as occurred during the one-day storage period. The difference in corrosion produced between a seven- and a ten-day test were considered insignificant. Therefore, for the practical considerations of stand utilization with this technique, the standard storage times established were one day and seven days. It is believed that the one-day test adequately determines the ability of the lubricant to protect the axle assembly against corrosion. However, a somewhat more precise determination can be made by utilizing the seven-day test, since the difficulties of visual rating are minimized by the increase in intensity of the corrosion formed.

(5) Sand Blasting of the Test Cover Plate

As is nearly universally mentioned in the literature on the development of corrosion test technique the preparation of the corrosion sample is of utmost importance. Very early in the development of this technique detailed attention was given to a preparation of the cover plate of the axle. It had been determined from field observations that the greatest amount of corrosion occurred on the cover plate. This was probably due to the fact that during periods of falling temperature water would preferentially condense on this cooler surface in the axle. In addition, the cover plate is formed from a low carbon steel rather than the high alloy, corrosion resistant materials used on most of the rest of the axle parts. Since the cover plate of the production axle used in this technique was formed by drawing, it would be expected that the surface would have a relatively heavy oxide coat. In accordance with standard techniques for other corrosion tests, it was specified that the cover plate for the test axle would be sand blasted prior to installation on the axle. In order to avoid contaminating the cover plate after sand blasting, it was further specified that loose sand should be washed from the plate by Stoddard solvent and that the cover plate be allowed to air dry without the use of external means.

Upon evaluating cover plates from other laboratories cooperating in the reproducibility study mentioned above, it was observed that the surface roughness formed by the sand blasting in the various laboratories differed. Upon investigation it was determined that this difference was due to the use of silica of varying size and from different origins. In addition, the velocity of the air and the pressure used in the blaster had been left to discretion of the operating laboratories. The specification was then tightened to require silica of a certain hardness and grain size and to specify the pressure to be used in the sand blaster. It is believed with the present specifications in the test technique that adequate control now exists upon the surface condition of the cover plates being used in this test.

III. PHASE II - STUDY OF DYNAMIC LOADING OF AUTOMOTIVE HYPOID GEARS

A. Introduction

The study of dynamic loading in automotive hypoid gears was conducted in two parts. Part one was concerned with the determination of dynamic torques applied to ring and pinion gears under the conditions of several standard full scale gear lubricant test procedures. In this part of the work dynamic torque measurements were made using strain gage instrumentation in both laboratory and field conducted gear lubricant tests. The second part of the work was concerned with the determination of dynamic loading on hypoid gears in Army trucks as operated by the Army Ordnance Test Unit on summer gear lubricant field test programs at both the Yuma Test Station and Death Valley, California. These two parts of this phase of the contract are discussed separately below.

B. Study of Dynamic Loading in Standard Gear Lubricant Tests

1. Object

At the initiation of this contract, there was considerable concern in the petroleum and automotive industries as to the lack of repeatability and reproducibility of the CRC L-19 High Speed Gear Lubricant Test Procedure. It was believed that this lack of uniformity in test results was probably caused by one or both of two factors: the lack of uniformity in production test gears, and the lack of uniformity in the loads being applied to the test gears during the tests.

It was apparent that reliable information on the actual torque being applied to the ring and pinion gears in the L-19 test would be required before adequate control measures could be taken. In order to provide empirical data on the torques of standard full scale gear lubricant tests, this program was undertaken.

2. Method

The determination of dynamic torque loading of rear axle gears in standard gear lubricant tests was accomplished by recording axle torque, sensed by electric resistance strain gages, on a high speed light beam type oscilloscope. Since all of the work was carried out on a 1949 Chevrolet automobile drive assembly, it was possible to use two methods of strain gage application to measure the drive system torque. One method involved the application of the electric resistance strain gages in a bridge network fastened to the axle shaft of the automobile drive system as shown in Fig. 13. The signal was obtained from the strain gage bridge through a slip ring assembly. By mounting the gages with their centerlines perpendicular to the two adjacent gages the effect of bending in the axle shaft was eliminated. The use of four active gages in the bridge produced four times

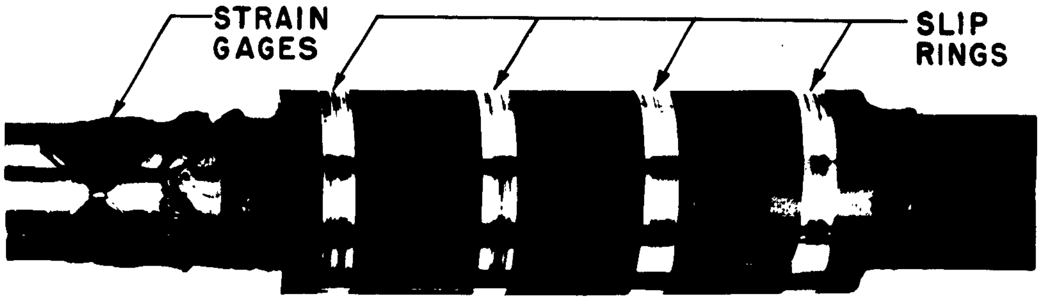


FIG. 13 — STRAIN GAGES AND SLIP RING ASSEMBLY
ON AUTOMOBILE AXLE SHAFT

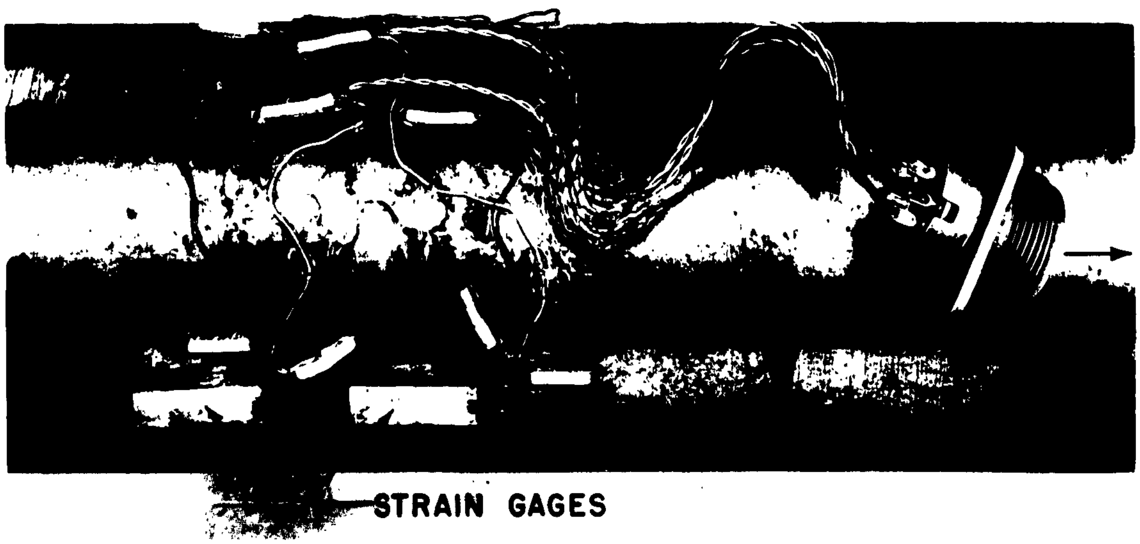


FIG. 14 — STRAIN GAGES ON AUTOMOBILE TORQUE TUBE

the signal obtainable with one gage.

Since the Chevrolet drive system employed on the 1959 vehicle was a torque tube drive in which the axle housing is restrained from rotation by the propeller shaft housing, the bending force in the propeller housing was proportional to the torque on the gears. By mounting the four strain gages on the opposite sides of the propeller shaft housing, it was possible to measure the torque applied to the axles by determining deflection of the housing itself. Figure 14 is an illustration of the mounting of strain gages on the Chevrolet torque tube.

The dynamic torque applied to the system was measured by impressing upon the strain gage bridge a three KC alternating carrier wave. The distortion of the strain gages in the torque tube bridge or the axle bridge produced an amplitude modulation of this signal. The signal was transferred to an AC amplifier where it was amplified and demodulated and then applied to a light-beam type oscilloscope. The oscilloscope trace then showed the torque applied to the axle system as a graphic presentation on photo-sensitive paper.

3. Results and Discussion

a. CRC L-19 Test

Torque recordings were made of several cycles of CRC L-19 high speed gear lubricant test technique as conducted in the laboratory on a T dynamometer stand and as conducted in an automobile on the highway. This test procedure involves two parts, both consisting of full throttle accelerations alternated with closed throttle decelerations. The first part of the procedure consists of a high gear acceleration from 10 mph to 40 mph, a sudden closing of the throttle and a deceleration to 10 mph. At that point the throttle is again opened suddenly and the vehicle or rig is accelerated to 40 mph. Five accelerations and decelerations complete this sequence of the test. The second part of the test employs full throttle acceleration from 60 mph to 80 mph and a closed throttle deceleration back to 60 mph. Then the throttle is opened suddenly and again a full throttle acceleration to 80 mph is employed. Ten accelerations and decelerations complete this sequence of the test.

The resultant loadings on the pinion gears of the test axle are alternating drive and coast shocks. The peak torque values obtained from the instrumented tests conducted in the laboratory and on the highway are tabulated in Appendix B. In order to make these values comparable to shock loadings on other rings and pinions, the torque values have been reduced to the form of pounds-load on the gear tooth per inch of gear face. This parameter is often used as a rough indication of the actual unit loading on the gear teeth. There is no known method at this time for computing actual unit loadings existing on hypoid gear teeth taking into account elastic deformation of the teeth.

Figure 15 presents the comparison of peak ring gear loads applied to the L-19 in both the laboratory and road tests. It may be observed that there are appreciable variations in peak load from cycle to cycle in either the laboratory or road test; but from the data available, it would appear that there are generally greater variations in the road test loadings than in the laboratory test loadings. This is most noticeable during the portions of the test at which drive side shocks are applied (at 10 mph and 60 mph).

The greatest differences between shock loadings applied to the gears in the laboratory and road tests were in the drive side shocks applied upon full acceleration at 10 mph. At that point it is not possible to apply the same rate of throttle opening in the laboratory as it is on the road

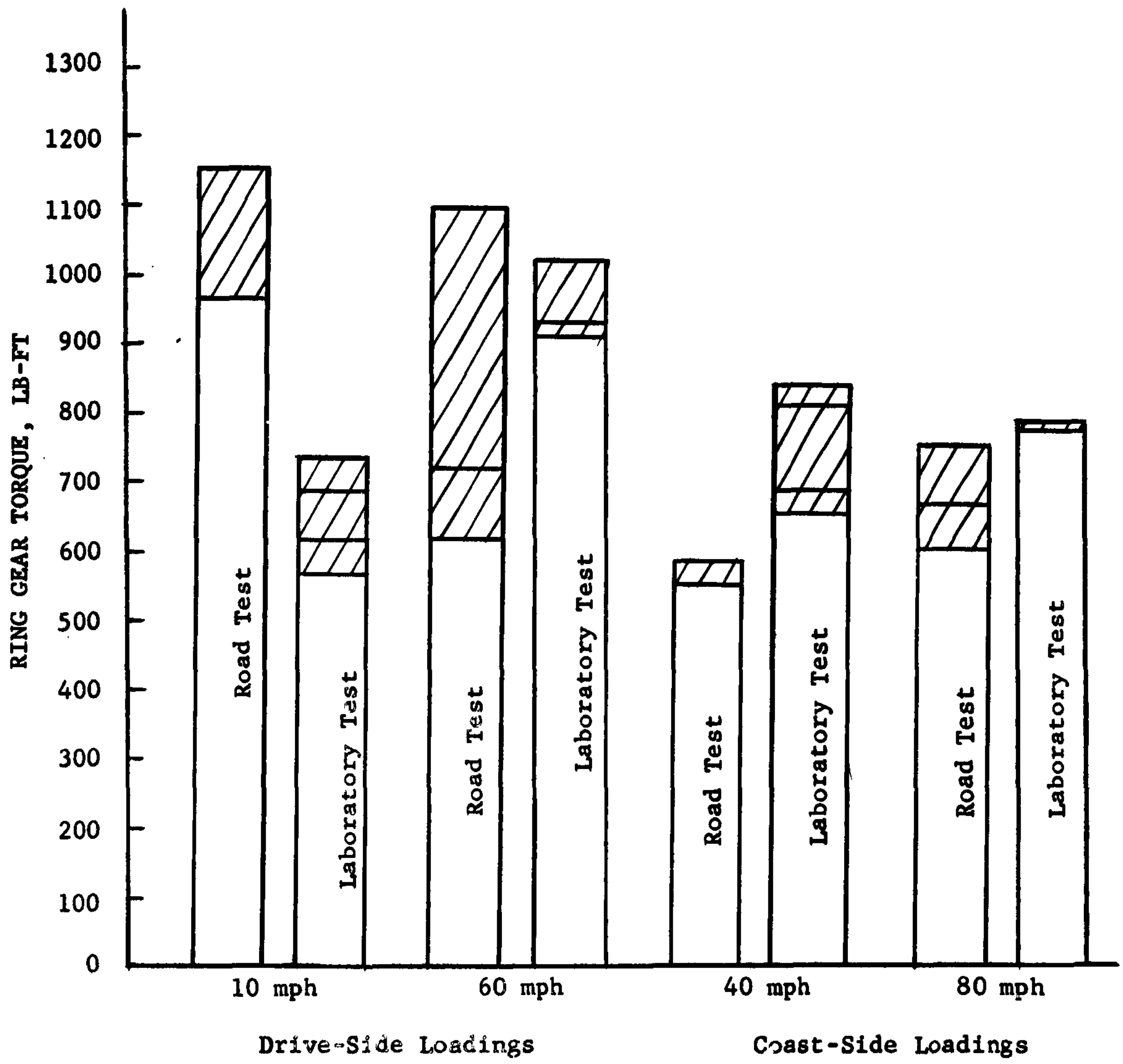


Fig. 15 COMPARISON OF PEAK RING GEAR TORQUES IN ROAD AND LABORATORY L-19 TESTS

because of severe bucking that occurs in the laboratory equipment at the low speed due to the rigid drive system coupling the engine to the large masses of the dynamometer armatures. Experience has dictated that in the interest of greater uniformity of loading at full throttle acceleration at 10 mph a retarded rate of throttle opening is required in the laboratory.

It may be noted that in the laboratory test, the coast side shock at 40 mph is greater than the drive side shock at 10 mph. In gear lubricant qualification tests run in the laboratory over a period of years, it had been noted that if a lubricant failed during the 10 to 40 mph cycles and the failure was only on one side of the teeth, it would almost invariably appear on the coast side of the teeth. The higher load at 40 mph along with the increased rubbing velocity of the teeth tends to explain this observation. Failures in the laboratory L-19 test during the 60 to 80 mph cycles have never been decisively determined to be initiated on either the coast or drive side of the gears. This might be expected from observing the laboratory torque data since the load applied to the drive side at 60 mph is higher than the load applied to the coast side of the teeth at 80 mph but the rubbing velocity is obviously higher at the higher speed. It would appear that these two factors tend to balance each other so that the severity of lubrication requirement at the 60 and 80 mph shock loadings are roughly equivalent.

During the early work on the recording of the torque values from the L-19 test as run in the laboratory, the engine was operated on an acceleration sequence with spark plugs which were intermittently fouling out. Figure 16 shows a trace obtained from this work. It will be observed that on several occasions during this acceleration torque values equivalent to the peak torque obtained in a normal acceleration were obtained on the

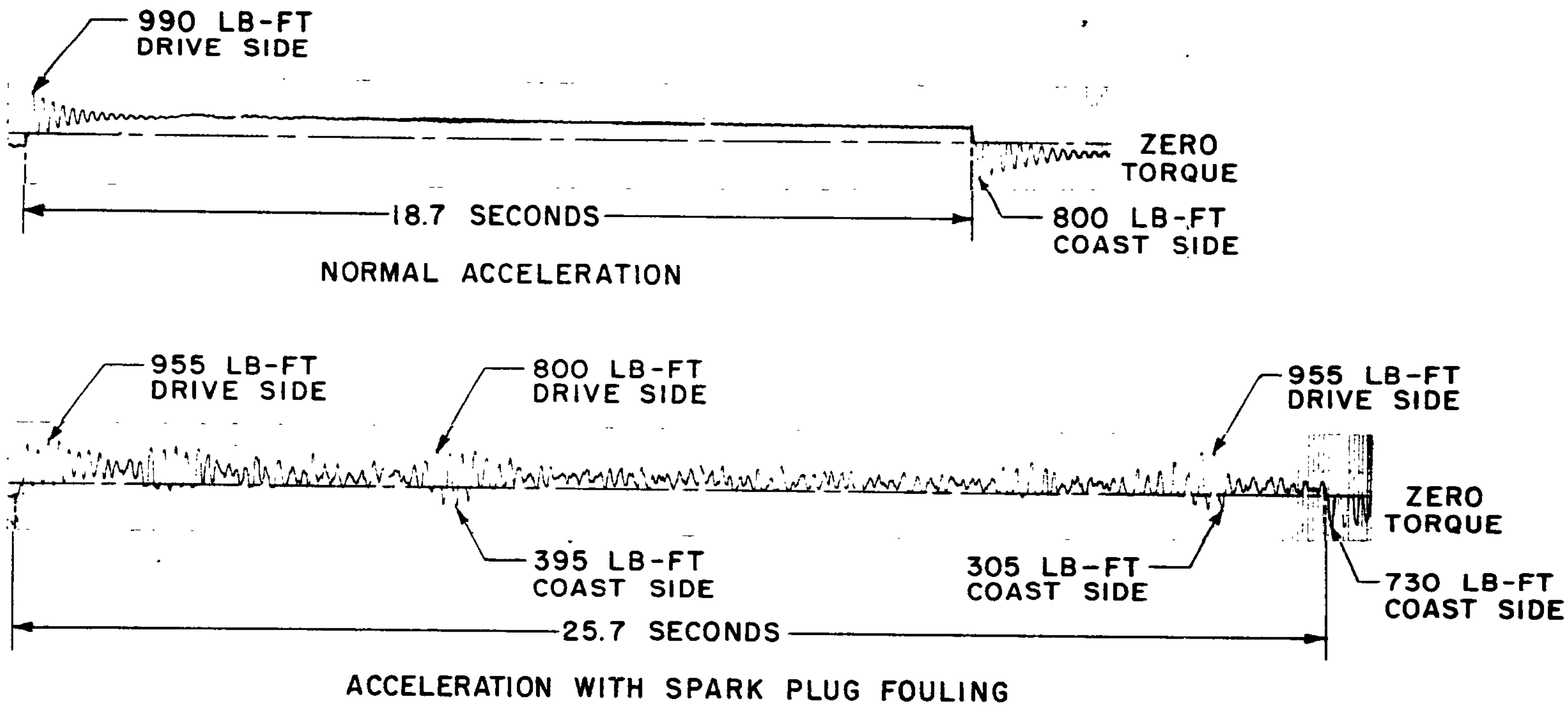


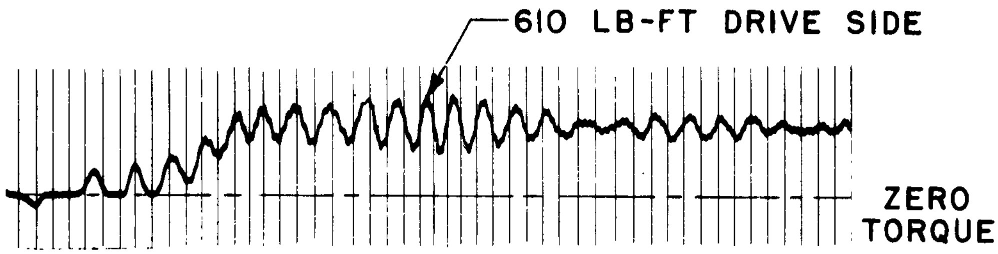
FIG. 16 — EFFECT OF SPARK PLUG FOULING IN LABORATORY L-19 TEST

drive side of the gears. In addition, on two occasions the loading actually reversed to be applied on the coast side of the gears. Other than the fact that the acceleration took approximately seven seconds longer than a normal acceleration, there was no visual evidence of any malfunction in the test equipment. It is quite likely that without torque recording equipment on this test it would have been assumed that the loading was less severe than normal because of the increased acceleration time. However, it is obvious from the torque record that the loading was considerably more severe than normal. This case emphasized the importance of obtaining accurate torque information on any shock loading procedure for testing of gear lubricants.

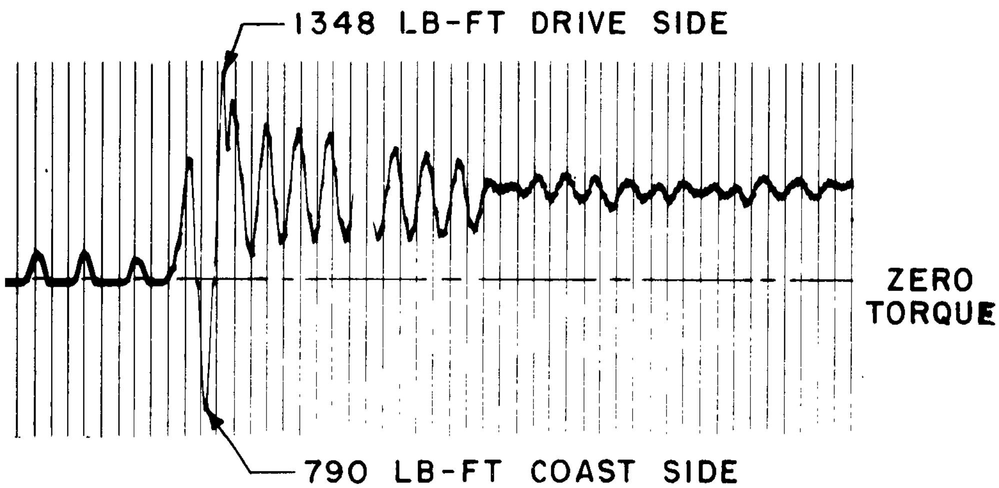
Further experimentation was conducted to determine whether the shock loadings applied to the gears could be varied appreciably by using other means of throttle control. Figure 17 is the torque record obtained from an L-19 acceleration at 10 mph using an electric solenoid throttle control system instead of the standard hydraulic manually-operated throttle control system. A drive side loan of 1348 lb-ft was produced on the gear teeth. This is to be compared with the average loading of approximately 610 lb-ft obtained using the hydraulically operated throttle. The coast side loading on the gears was not appreciably affected by more rapidly closing the throttle. It is apparent that the drive side loading on shock type procedures can be selected and controlled depending upon the rapidity with which the throttle is actuated.

b. GM Shock Test

Several recordings were made of shocks in both parts of the GM shock test procedure. This procedure also consists of two main parts. The first part includes acceleration to 70 mph, and engine declutching, the ignition is turned off and the car allowed to coast to 65 mph. Upon reaching



HYDRAULIC THROTTLE CONTROLS



ELECTRIC SOLENOID THROTTLE CONTROLS

FIG. 17 – EFFECT OF THROTTLE ACTUATION RATE
IN LABORATORY L-19 TEST.
(THROTTLE OPENED SUDDENLY AT 10 MPH)

that speed, the clutch is snapped out and the car allowed to coast down to 40 mph. This cycle is repeated seven times. The second part of the test is accomplished by accelerating in second gear to 50 mph, declutching, turning off the ignition, coasting to 45 mph and snapping the clutch into engagement. The car is allowed to coast to 20 mph. This cycle is repeated five times. Figure 18 shows the comparative peak load values obtained in the L-19 and the GM shock tests. It may be seen that in the GM shock test the high gear shock at 65 mph on the coast side of the gear was approximately three times as great at the largest coast-side shock load in the L-19 test. This shock was approximately twice as large as the heaviest shock in the road L-19 test (at 10 mph on the drive side of the gears). It may be observed from the photographs of the oscillograph traces from these shocks in Fig. 19 that after the initial heavy shock the torque oscillates for a period of about one-half second between 420 lb-ft and 1330 lb-ft. In other words, the load on the gears remained in the range of the maximum L-19 shocks for about one-half second, normally time for more than 25 revolutions of the pinion. From observing the skid marks on the highway after such a shock is applied to the drive system, it is apparent that the direct application of the dead engine to the drive system essentially stops the entire drive system until the inertia of the automobile, through the application of load from the tires through the axle and propeller shaft forces the engine to turn. It would therefore appear that this very high oscillating load which is applied to the gears for approximately one-half second may be effective on only a few of the gear teeth.

The shock torques experienced in second gear at 45 mph in the GM shock test were approximately six times as great as the largest coast-side shock recorded in the road L-19 test and were roughly four times as large as

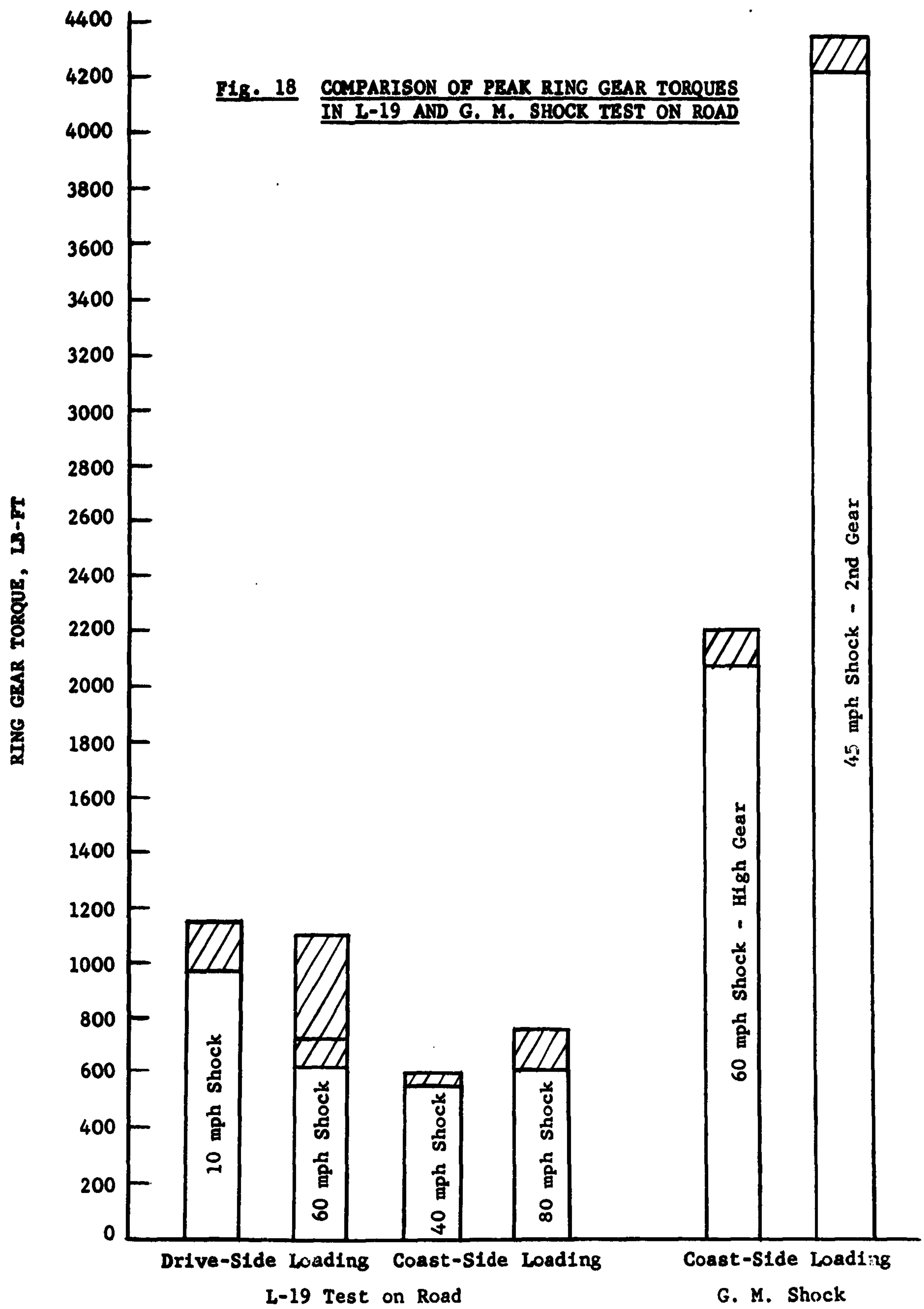


Fig. 18 COMPARISON OF PEAK RING GEAR TORQUES IN L-19 AND G. M. SHOCK TEST ON ROAD
RING GEAR TORQUE, LB-FT
4400
4200
4000
3800
3600
3400
3200
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
10 mph Shock
60 mph Shock
40 mph Shock
80 mph Shock
60 mph Shock - High Gear
45 mph Shock - 2nd Gear
Drive-Side Loading
Coast-Side Loading
L-19 Test on Road
Coast-Side Loading
G. M. Shock

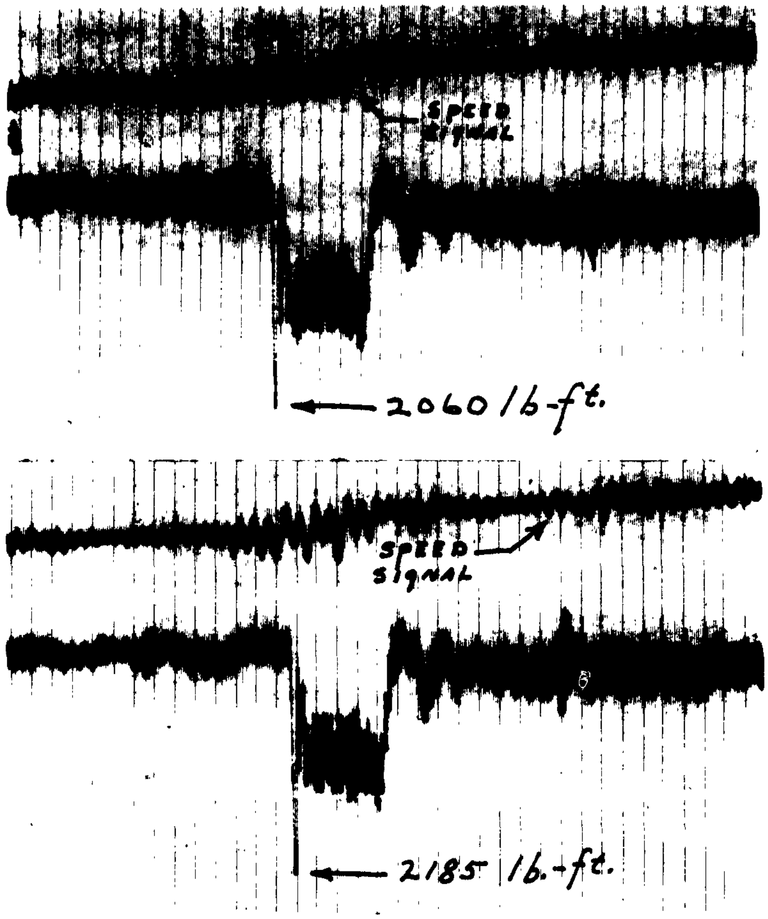


Fig. 19 COAST SIDE SHOCK LOADS AT 65 MPH IN HIGH GEAR ROAD GM SHOCK TEST

the drive-side shock loads in that test. As may be seen from Fig. 20, after the initial shock the torque oscillates for approximately 0.6 seconds from nearly 0 torque in some cases to loads equalling the original shock pulse. From observing these shock loadings it can be seem why the GM shock test presents such a severe lubrication requirement.

4. Conclusions

From the results of these studies the following conclusions have been drawn.

a. L-19 Test

(1) The maximum shock loads in both the road and laboratory L-19 tests were found to vary considerably from cycle to cycle.

(2) For the accurate determination of actual loadings on the gear teeth during the test and for the monitoring of engine and drive system condition, the value of dynamic torque measurement systems has been amply indicated by this study.

(3) The laboratory L-19 test appeared to be somewhat more repeatable with respect to peak shock loads and characteristics of shock loading than the road test.

(4) With the exception of the drive side shock at 10 mph, there did not appear to be appreciably significant differences between the shock loadings in the road and laboratory tests.

b. GM Shock Test

(1) The high gear shock at 65 mph in the road GM shock test produced gear torques approximately three times as great as the maximum coast side torques and approximately two times as great as the maximum drive side torques in the road L-19 test.

(2) The second gear shock at 45 mph in the GM shock test produced coast side gear torques approximately six times as large as the maximum coast side torques and approximately four times as large as the maximum drive side torques in the road L-19 test.

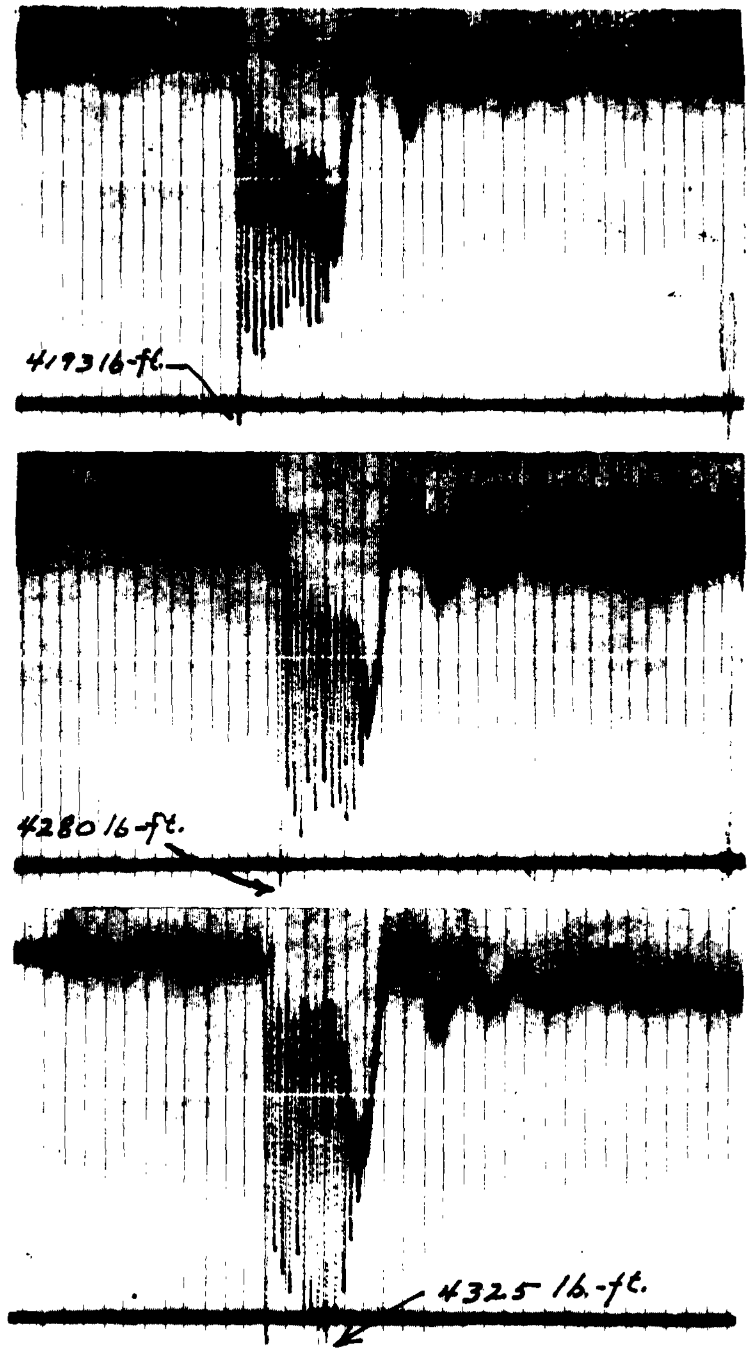


Fig. 20 COAST SIDE SHOCK LOADS AT 45 MPH IN SECOND GEAR ROAD GM SHOCK TEST

C. Study of Dynamic Loading in Army Field Test at Yuma, Arizona

1. Object

For several years prior to this study a series of tests had been run each summer at the Yuma Test Station by the Army Ordnance Corps. These tests had been instrumental in pointing out the inadequacy of present military specification gear lubricants for extremely severe operating conditions in military trucks. The tests had also proved valuable in proof-testing new experimental gear lubricant products.

Because of the value of the Yuma test work it was considered highly desirable to measure the dynamic gear loadings being applied to the vehicle gears in the course of the prescribed tests. It was also desired to determine whether any dynamic loading on the gear teeth were significant. It was expected that shock loadings might have been applied by the oscillation of the heavily loaded trailers or during the shifting of gears.

2. Method

Electric strain gage instrumentation was applied to the axles of the Army 2-1/2 ton M-211 truck and the Army 3/4 ton M-37 truck. The M-37 truck was loaded to 100 per cent of its net highway load rating of 2,000 lbs. The gross weight of the M-37 truck during this study including the pay load, test equipment, and the three-man crew was 8,915 lbs. The M-101 trailer attached to the M-37 truck was loaded to approximately its rated road load of 3,500 lbs gross. However, this was less than the rated towed load rating for the M-37 truck of 6,000 lbs for highway operation. The M-211 truck was also loaded to 100 per cent of its highway net load rating of 10,000 lbs. The gross weight of this vehicle during this study including the pay load, the test equipment and the three-man crew totaled 24,835 lbs. The M-105 trailer which was towed by the M-211 truck was loaded to approximately 7,900 lbs gross. This was less than the rated towed load for the M-211 truck but more than the rating for the M-105 trailer. The M-211 truck is rated for 10,000 lbs of towed load under highway operation. A photograph of the instrumentation in place in the M-37 truck is shown in Fig. 21. Load measurements were made by the use of electric resistance strain gages mounted on the rear axle of the M-37 truck and the drawbar of its trailer. Load measurements on the M-211 truck were made of both the intermediate and rear axles and the drawbar of its trailer. For both trucks the speed of the vehicle was simultaneously recorded on the oscillograph record with the torque signals.

The test course used in these investigations was the same as that used during the summer tests of 1955 and 1956. In the Yuma area the course extended from a spot approximately five miles south of the Yuma Test Station on Arizona Highway No. 95 to a turn around spot approximately 35 miles north of the Yuma Test Station. The black top highway is layed across the desert with a minimum of grading resulting in many dips and rises through ravines. At the north end of the course, there is a hill approximately one mile long which required down-shifting of the transmissions on all vehicles. The highway has an average upgrade of 0.8 per cent in a northerly direction. The average speed for the M-37 vehicle when travelling northward was approximately 42 to 44 mph. When travelling southward the vehicle speed was approximately 46 to 48 mph. The average speed of the M-211 truck when travelling northward was approximately 1 to 2 mph slower than the M-37. When travelling southward the M-211 was approximately 1 to 2 mph faster than the M-37. In both vehicles the course was traversed under full throttle except in passing through a few severe dips. Except for the starting and stopping at the two ends of the Yuma area course, the M-37

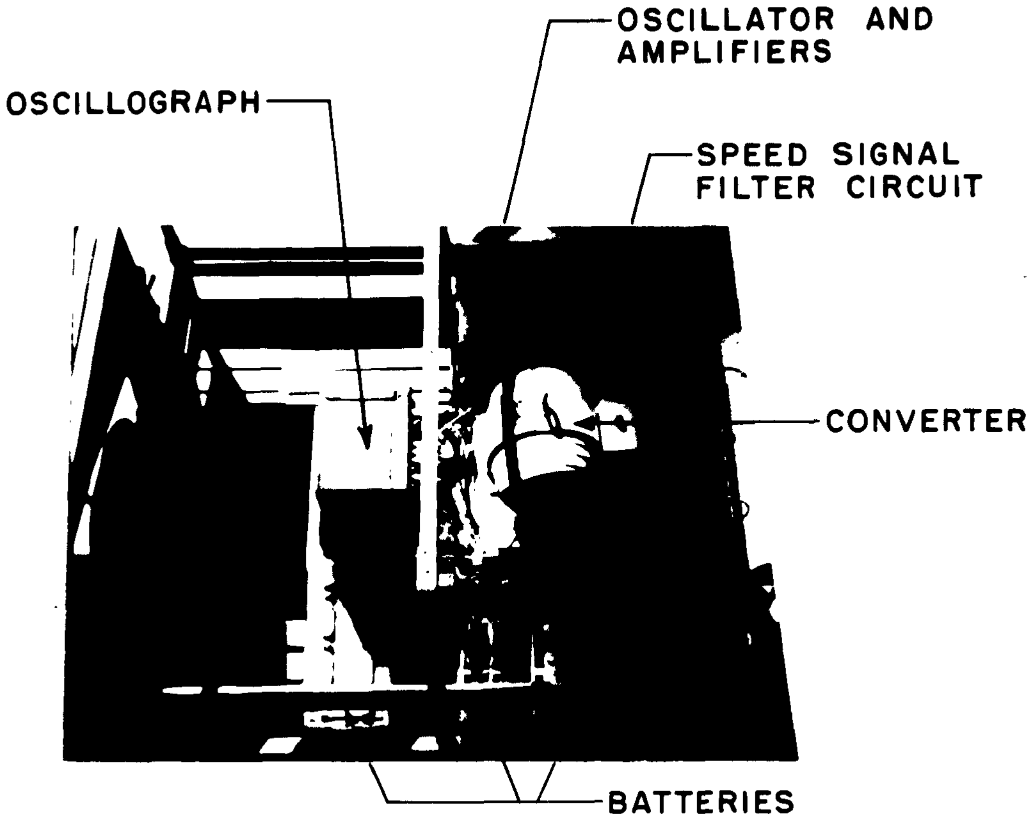


FIG. 21 – ARRANGEMENT OF INSTRUMENTATION IN M37 TRUCK

truck usually shifted gears only once in the 40 miles. The M-211 truck shifted gears a number of times during its travel northward on the Yuma area test course. These would usually occur as it was going into a dip or coming up the short grade on the other side. When travelling southward the M-211 shifted very infrequently. During the summer tests approximately 6,000 miles of operation were obtained over this portion of the test course in both vehicles.

In the Death Valley area the vehicles daily traversed a course which started on the east side of the Valley, crossed Daylight Pass, descended into the Valley, crossed the floor of the Valley, ascended Towne Pass, and descended on the other side to Panamite Springs, California. At this point the vehicles turned around, climbed Towne Pass again, descended into the floor of the Valley, proceeded south on the floor of the Valley to Furnace Creek and turned around. They travelled back up the floor of the Valley to Daylight Pass, climbed the pass, and returned to their starting point in Beatty, Nevada. The total daily circuit was approximately 165 miles long. Of this distance approximately 50 miles were travelled on level road and the remaining 115 miles were spent in climbing up and down the passes. Since the 50 miles travelled on the floor of Death Valley was very similar to the operation on Arizona Highway No. 95 near the Yuma Test Station, it was decided that the most significant information would be obtained on the slopes of the two passes in Death Valley. When climbing the grades in Death Valley the M-37 vehicle maintained an average speed of approximately 16 mph. On one portion of the slope of Towne Pass, the truck could maintain no more than 8 mph. The M-211 vehicle maintained an average speed of 13 to 15 mph on the slopes in Death Valley. On the steepest portion of the slopes the M-211 was also slowed to approximately 8 mph.

The slopes in Death Valley were surveyed to determine the points of maximum grade on both Towne and Daylight Passes. The torques produced on these grades were recorded in both vehicles. In addition, an attempt was made to locate slopes which were approximately equal to the average slopes of both Towne and Daylight Passes. The axle torques were also recorded over these portions of the road. Some gear shifts were recorded on other portions of the hills in Death Valley.

3. Results and Discussion

Several environmental and operational conditions in these tests resulted in severe dynamic loading on the rear axle gears. As would be expected an increase in slope of the road and the addition of trailed loads adversely affected the gear loads. However, it was found that the largest loading upon the drive gears of the trucks was produced during transmission gear shifting. The effects of these three factors upon gear torques will be discussed separately in the following sections.

a. Effect of Road Load

Average ring gear torque values are tabulated for both the M-37 and M-211 vehicles in Tables C-1 through C-4 in Appendix C. Figure 22 shows an over-all comparison of representative values of ring gear torque from both trucks on the maximum slopes in Death Valley, the average slopes in Death Valley, the level road load conditions and the hill at the north end of the Yuma course. On the steepest grades in Death Valley the axle torque of the M-37 was about 13 times as great as when operating on a level road at a constant speed of 40 mph. In the M-211 vehicle, the gear loads were increased by 15 and 24-fold, respectively for the intermediate and rear axles over their level road loads at 40 mph. Less steep grades, of course, produced smaller load increases. However, the least loads recorded during

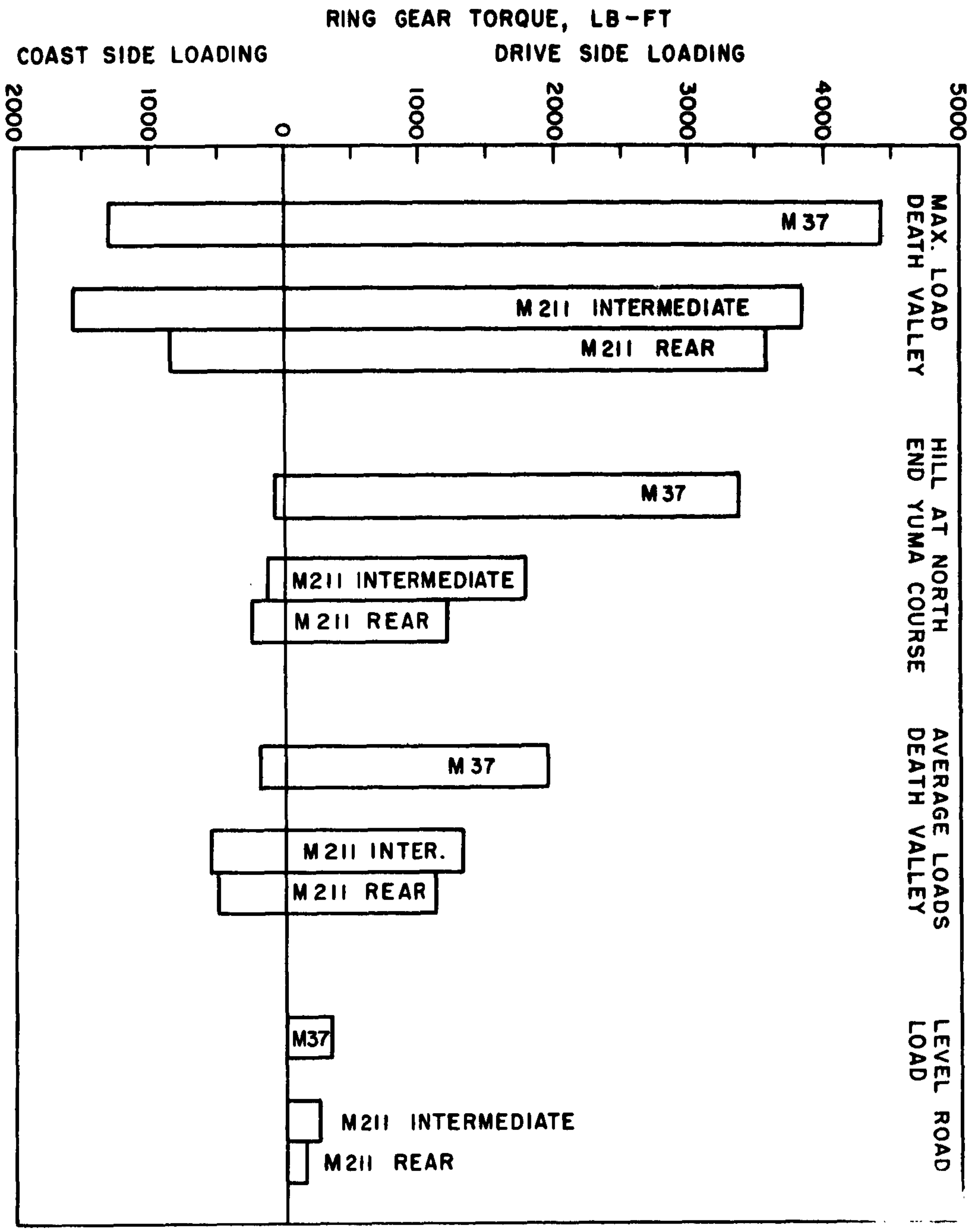


FIG. 22 – EFFECT OF GRADES ON TRUCK AXLE TORQUES
(YUMA GEAR LUBRICANT TEST COURSE)

hill climbing were approximately 5 to 7 times as great as the loads on corresponding gears under the road load conditions. Obviously, the loading on the coast side of the gears produced by the drag of the engine during descent of the grades, was considerably less than the drive side loading required to climb the same slopes. The unusually small coast side loading recorded on descending the hill at the north end of the Yuma course probably resulted from the fact that the hill is short and the drivers descended at high speed. In most cases the engine throttle was held part-way open during this descent.

It will be observed that the loading on the M-211 axles are less than those on the M-37 rear axle in comparable parts of the course. The fact that the M-211 truck has two axles certainly accounts for a large portion of this difference. It would seem apparent that the lubrication requirement for the M-37 gears is greater than that of the M-211 truck gears. The results of the summer Yuma gear lubricant test would tend to substantiate this conclusion. The lubrication failures on the M-37 gears have been more frequent and more severe than those of the M-211 gears.

The torque values from the M-211 truck compared in Fig. 22 are adjusted to be equal to the values which would have been obtained when towing a 7,825 lb trailer. In the actual experimental work the trailer was loaded only to 5,500 lbs. However, in subsequent work near the Yuma Test Station, a correction factor was determined by measuring the increase in axle load resulting from the increased trailer load. In Table C-4, it will be observed that the torque on the intermediate and rear axle gears of the M-211 truck, when climbing the maximum slope on the west side of Towne Pass, appeared to be out of line with the rest of the data presented in the table. It should be noted that the truck on this slope was in low-third gear, whereas in climbing the maximum slope on Daylight Pass, the truck was in low-second gear. Since the truck was travelling at the same speed on both of the slopes, it would be expected that the torque to the rear axle when operating in low-third would be considerably less than when operating in low-second, due to the difference in gear ratio and to the different positions on the engine torque curve.

Figure 23 presents a comparison of the average loads obtained on the M-37 truck in this program in both Death Valley and the Yuma area with the loads applied during the CRC L-37 High-Speed Low-Torque and High-Torque Low-Speed Gear Lubricant Test Procedure. Since the L-37 test procedure was developed to correlate with the results of the Yuma tests, it is of interest to compare the torque values obtained in the two tests. The L-37 test is composed of two portions similar to the two parts of the Yuma test. The initial portion of the L-37 test is run at relatively high speed and low torque. The second portion of the L-37 test is operation at relatively high torque and low speed. In the Yuma area, the M-37 vehicles were operated over Arizona Highway No. 95 at nearly top speed and correspondingly at relatively low torque. In the high speed portion of the field test, the M-37 trucks travelled at an average speed of approximately 45 mph. The corresponding axle speed is 455 rpm. In the L-37 procedure the initial portion of the test is conducted at 440 $\pm$ 5 rpm. It may be seen, therefore, that the speeds of these two procedures correspond quite closely. Because of the wide variety of operating conditions on the gears in the field test in the Yuma area, it is impossible to obtain one single load value which is equivalent to the average loading on the gears. However, Fig. 23 presents the representative loads at various portions of the Yuma area course. It may be observed that on climbing the hill at the north end of the course

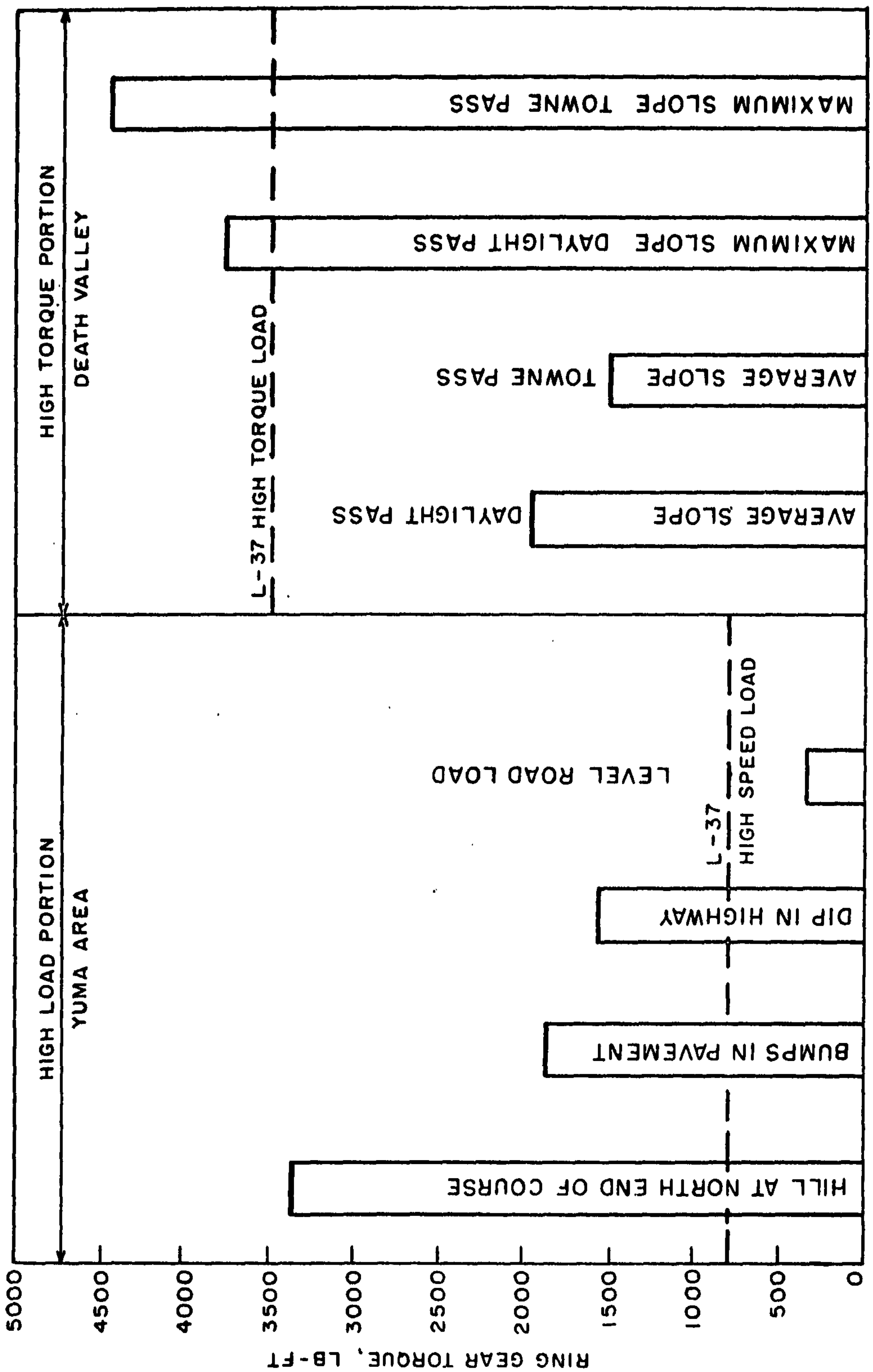


FIG. 23 - TYPICAL GEAR TORQUES IN M37 TRUCK DURING FIELD TESTS.

the loads are approximately 4 times as great as the steady loads in the initial portion of the L-37 test. The loads which result from dips or bumps in the highway are approximately 2 times as great as the steady loads in the L-37 test. However, the road load for the M-37 vehicle is less than half of the loading in the L-37 test. It would therefore be concluded that the 788 lb-ft of ring gear torque of the high speed portion of the L-37 test closely approximates the time-average of the ring gear torques during the high speed portion of the Yuma test.

Upon comparing the L-37 high torque gear loading to the loading on the M-37 truck in the Death Valley area, we see that the average torques on the M-37 gears in hill climbing are lower than the loading of the L-37 test procedure. However, the maximum steady loads produced on the gears in the Death Valley area are greater than the high torque loading in the L-37 test. It should be kept in mind that the steady gear loads, even on the maximum slopes in Death Valley, were not the highest loads applied to the gears during the field test. The greatest loads were obtained under the shocks imposed by gear shifting. Again, it is not possible to determine one single representative average load for the Death Valley operation of the M-37 truck. However, it appears that the loading selected for the L-37 test is a reasonable value of loading to correlate with the Yuma field test data.

The axle speed during the high torque portion of the L-37 test is 80 rpm. The axle speed on the average slopes in Death Valley was approximately 160 rpm. On the maximum slopes in Death Valley the speed was approximately 86 rpm. It would therefore appear that in the Death Valley portion of the field test the axle speed is significantly greater than the speed selected for the high torque portion of the L-37 test.

b. <u>The Effect of Trailed Loads</u>

In the series of tests conducted by the Ordnance Test Unit, it was found that the addition of heavily loaded trailers to Army vehicles caused a large increase in the lubrication requirements of the gears. In several cases, lubricants which operated satisfactorily in Army trucks without trailers failed completely when heavily loaded trailers were coupled to the trucks. As a result, speculation has arisen as to the effect of towed loads on the dynamic loading of the rear axle gears. In the investigations of the gear loadings in the M-37 and M-211 Army trucks, the oscillograph records of axle torques and the trailer draw-bar pull were studied to determine whether oscillations were set up between the truck and the trailer under a variety of test conditions. It was theorized that if these oscillations were set up they would result in large shock loadings on the rear axle gears of the vehicles. Examination of the oscillograph records from this work has revealed very few cases in which the trailer overtook the towing truck or oscillated in the trailer hitch. However, the gear loads were increased by the addition of the trailer under all operating conditions. The addition of heavily loaded trailers to the trucks increased the peak torques resulting from gear shifting as well as the steady torques resulting from normal operation on roads ranging from level highway to steep grades.

Gear load measurements were made with the M-211 truck operating over the portion of the test course in the Yuma area without a trailer and also when towing trailers of two different weights. The trailer gross weights used were 7,825 lbs and 5,500 lbs. A tabulation of the average torque values obtained with the three trailed loads in given in Tables C-3 and C-4. The average per cent increase in gear torque load which occurred during hill climbing was determined and plotted in Fig. 24. It will be observed that the addition of the 5,500 lb trailer to the M-211 truck caused an increase of approximately 9 per cent in the gear load over the loading without a trailer. When the trailer weight was increased to 7,825 lbs the rear axle torque was 30 per cent greater than that recorded when no trailer was attached to the vehicle. It should be kept in mind that Fig. 24 is based upon a limited amount of data and an ultimate conclusion as to the effect of trailed loads should await further data on this phenomenon.

As might be expected the effect of trailed load appears most pronounced during hill climbing on either the steady load produced on the gears or on the shock loads produced during gear shifting. In level road operation very small increases in axle torque were observed with an increase in trailed load. Also in passing over dips and rises in the road as well as small bumps there was a slight increase of axle torque with an increase in trailed load. A trailer used for these studies is rated at approximately 7,900 lbs. However, the M-211 truck is rated for 10,000 lbs of towed load. It would therefore seem worthwhile to increase the trailed load to 10,000 lbs. and observe the effect on both axle torque and performance of reference gear lubricants.

With the 3,520 lb trailer attached to the M-37 vehicle, the ring gear load was increased an average of 38 per cent due to the trailed load. It is again apparent that the addition of a heavily loaded trailer to these Army vehicles can result in significantly increased ring gear torque over a variety of operating conditions.

A further effect of heavy trailed loads on Army trucks in these tests will be discussed in the succeeding section on gear shifting. The loading of the engine and drive system had a pronounced effect upon the

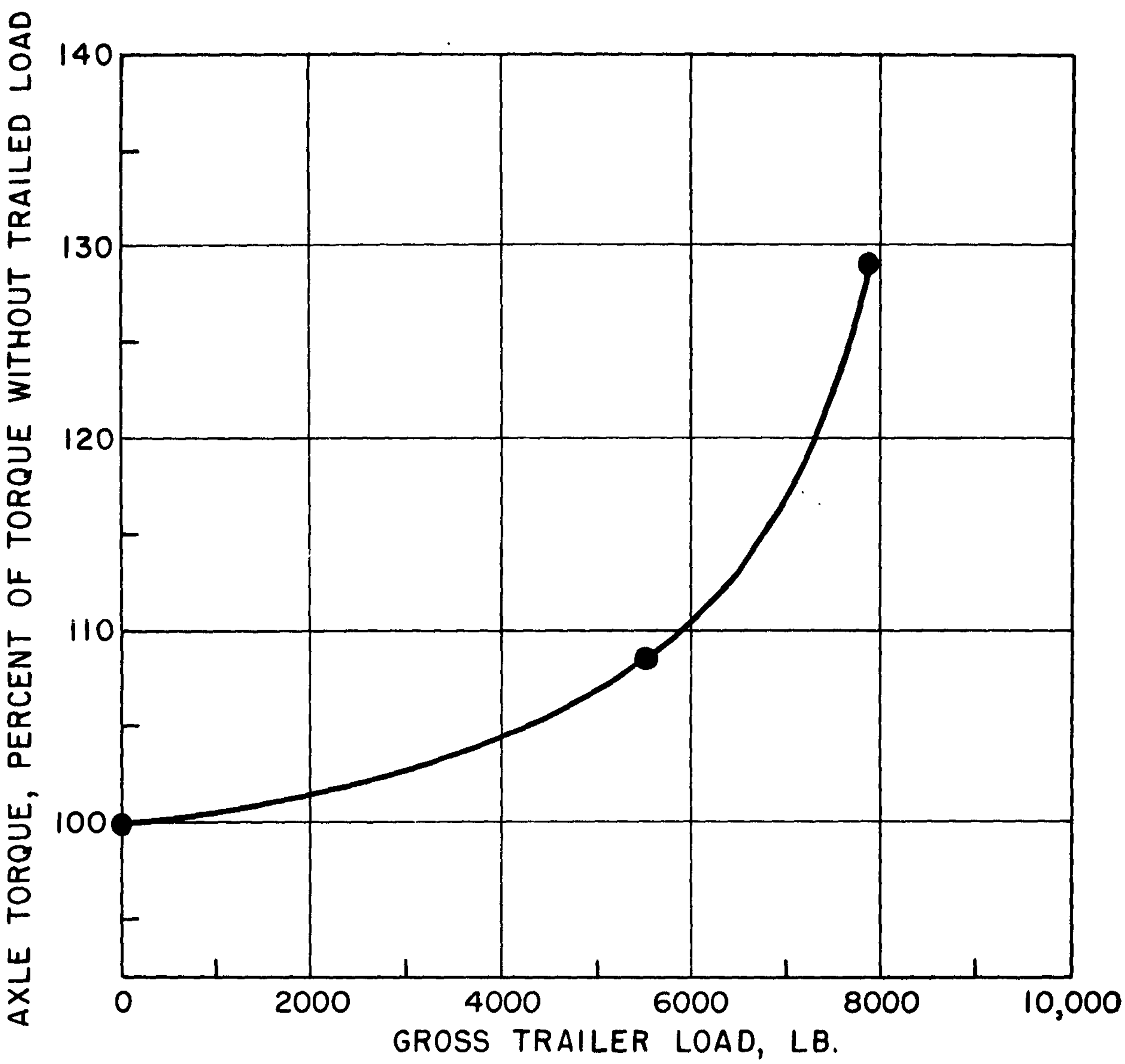


FIG. 24 — AVERAGE EFFECT OF TRAILED LOAD ON AXLE TORQUE IN HILL CLIMBING

amount of shifting required to traverse the grades. This was especially true in the M-211 truck with its automatic transmission.

c. <u>The Effect of Gear Shifting</u>

As was found in earlier project work on light automobiles, the peak loading which occurs due to gear shifting is the highest gear loading resulting from any normal operating condition. Tables C-5 and C-6 present average values obtained during a variety of gear shifting in the M-211 and the M-37 vehicles, respectively. The occurrence of gear shifting was most numerous in the M-211 truck which incorporates an automatic transmission. On some slopes in the Death Valley area, the transmission was continually shifting up or down in order to meet the load requirements of the vehicle on the slope. This was particularly true when operating on some of the steepest slopes in third gear of the low range. The vehicle would speed up due to an excess of power until it had reached the gear shifting speed. At this point, it would shift into fourth gear. However, fourth gear did not produce enough power at the rear axle to carry the load over the grade and the vehicle again slowed to the shifting speed and changed into third gear. Under certain conditions, this shifting would occur quite rapidly. On several occasions, shifts from low-third to low-fourth and back to low-third in a period from 1 to 1-1/2 seconds were recorded. On one occasion, the M-211 truck shifted between low-third and low-fourth gears 10 times in 47 seconds. On another occasion, this vehicle shifted between the same gears a total of 7 times in a little more than 32 seconds.

Figure 25 shows a typical oscillograph record of a shift in the M-211 truck from low-third to low-fourth and back to low-third. In this case, the two shifts were completed in approximately 1.4 seconds. The highest load obtained was on the drive side of the gears just after engage-

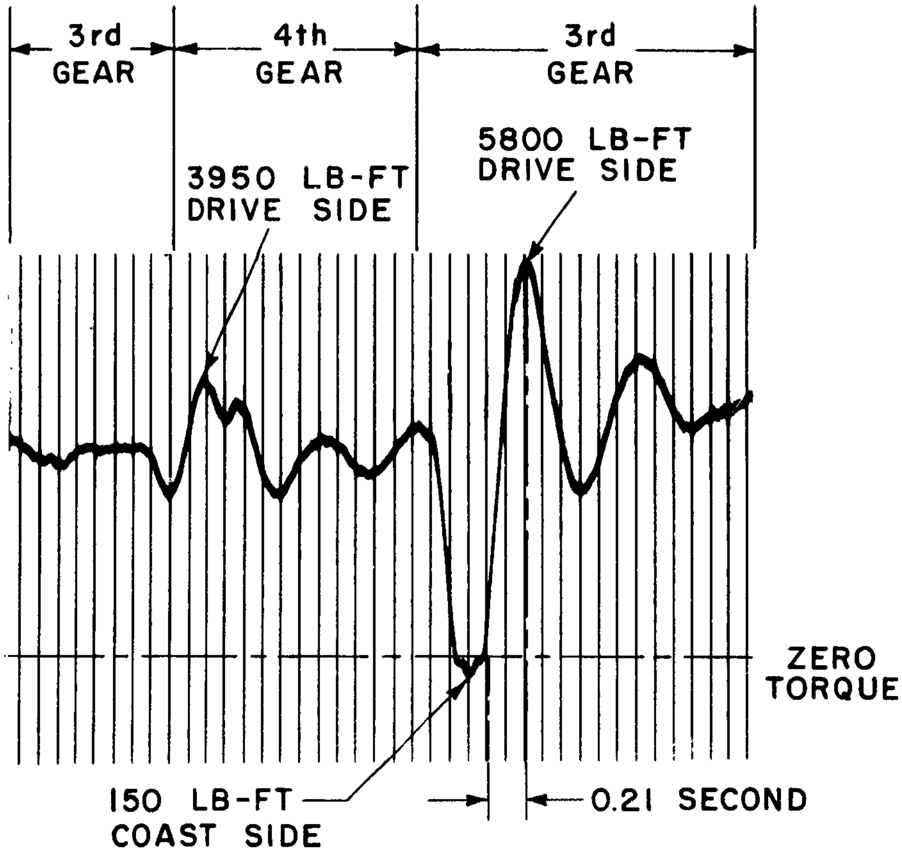


FIG. 25 – RING GEAR TORQUE IN M211 TRUCK DUE TO GEAR SHIFTING (OPERATING IN LOW RANGE)

ment in third gear. During this shift, the peak load amounted to approximately 5,800 lb-ft on the ring gear. On this same portion of the course, approximately 3,240 lb-ft of ring gear loading was required in order to climb the hill at a steady speed. Whereas the shift between low-third and low-fourth gear ranges was the most common one that occurred during the Death Valley work, there were numerous other shifts which were recorded at various times. Table C-5 lists the average of the maximum shock loads which occurred in the M-211 truck during the shifts designated. In high gear range, shifts were recorded between high-first and high-second, between high-second and high-third and between high-third and high-fourth. These gear shifts were not recorded under identical circumstances. It may be observed that some were under level road conditions and others were on uphill climbs. Nearly all of the shifts in the high gear range were recorded in the Yuma area. Loads produced on the gears during high gear shifts were all between 1,150 lb-ft and 3,060 lb-ft of ring gear torque. These values are in the same range as maximum loads obtained under steady speed and load conditions on the steepest slope in the Death Valley area. In the case of the shift from high-third to high-second while travelling uphill, shock loadings occurred on both coast and drive sides of the gears.

As would be expected, the shifts in the low range produced torque on the gears considerably higher than the shifts in high range. The average torque produced by a number of shifts between low-third and low-fourth was higher than any steady loads recorded on the maximum slopes in Death Valley. The highest loads occur when shifting from a higher to a lower gear.

The shifting between high and low ranges produced appreciable shocks on the rear axle gears. This was particularly true when shifting from high-second to low-fourth. Well over 5,000 lb-ft of ring gear torque was produced on both the rear and intermediate axles of the vehicle. Since this shock occurred on the coast side of the gears which had a relatively small amount of rubbing contact, it would probably be expected that this shift produced the most severe lubrication requirement upon the gear teeth experienced in the field work. Figure 26 is a photograph of the oscillograph trace of the torque from the rear axle during one of the high-second to low-fourth shifts. During this shift the load increased from 9 to 5,860 lb-ft of ring gear torque on the coast side in a period of 0.14 second. Since the loading on the coast side of the gears throughout the test was lighter and less frequent than it was on the drive side of the gears, it is expected that the thickness of extreme pressure film established on the coast side of the gears would be less than that on the drive side. A suddenly applied heavy load such as this would present a severe lubrication requirement on the EP film.

As was observed in the case of the steady loads on the M-211 truck, the loading on the intermediate axle was greater than the loading on the rear axle during gear shifting. The results of the Ordnance tests at the Yuma Test Station appear to verify these data. When there was an apparent difference between the condition of intermediate and rear axle gear surfaces, the intermediate axle gear showed the greater surface disturbance.

Some average peak shock torques from the M-37 truck are shown in Table C-b. The magnitude of the shocks produced on the rear axle gears in this vehicle during gear shifting was largely dependent upon the operator's skill and disposition. It would be expected that considerable variation in peak ring gear torque would result from operation by different drivers. The data presented in Table C-6 is the average of shocks produced on the M-37

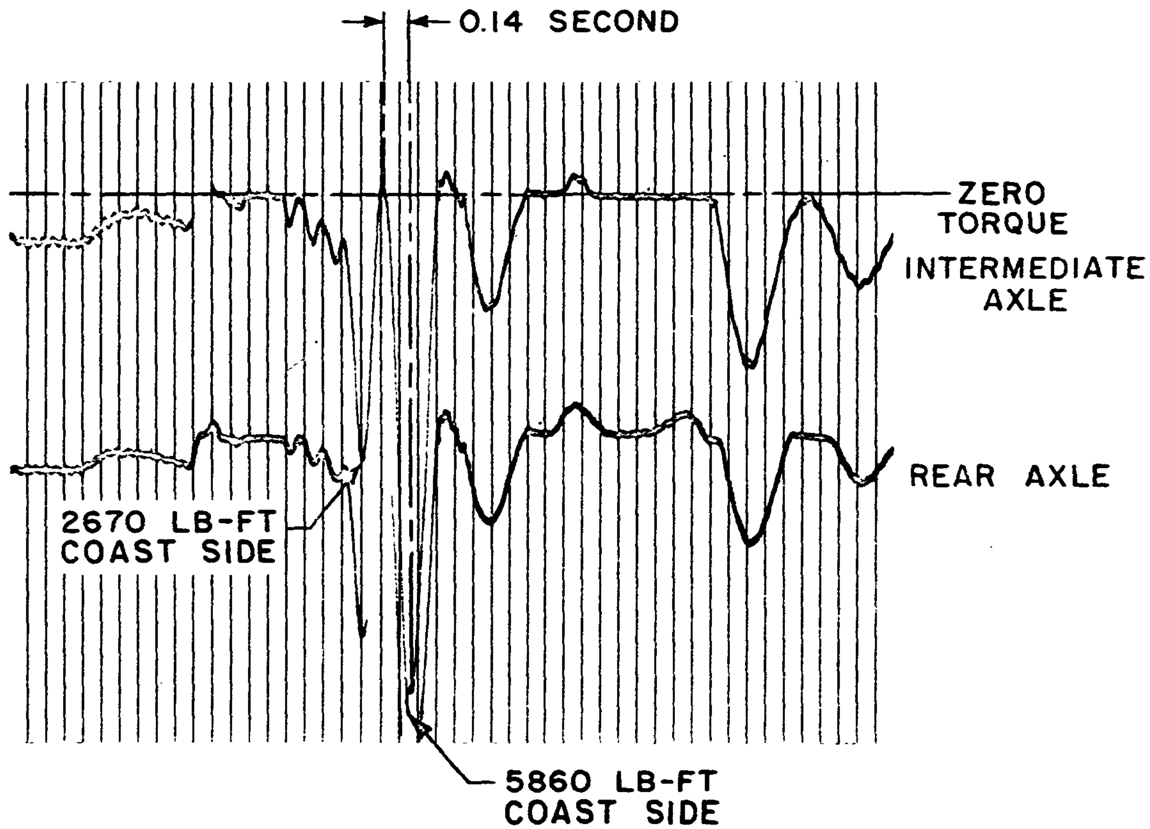


FIG. 26 — RING GEAR TORQUE IN M211 TRUCK DURING GEAR SHIFT FROM HIGH 2nd. TO LOW 4th.

truck when operated by two different drivers.

As in the M-211 truck, the largest gear loadings occurred when travelling uphill during the down shift. In most of the shifts which occurred in the M-37 truck, it will be noted that peak loadings occurred on both the drive and the coast side of the gears. When the shift was made while climbing a hill the initial load was on the coast side and, subsequently, on the drive side. When the shift was made while descending a hill the initial shock was on the drive side and the reaction shock on the coast side.

From the results of this project, it would be concluded that the shock loads produced by gear shifting would be a much larger factor in the M-211 truck than in the M-37 truck due to the difference in frequency of shifting of these two vehicles.

4. Conclusions

a. General

(1) The highest loads encountered in this program occurred during gear shifting.

(2) The M-37 truck gears generally carried higher torques than the M-211 truck gears in both mountain and level road operation.

(3) The addition of trailed loads caused large increases in gear torques especially in mountain operation. In the M-211 truck, gear loadings appeared to increase exponentially with an increase in the trailer gross load.

(4) With heavily loaded trucks, the climbing of steep grades increased the gear loading as much as 13 to 24 times that of level road load at steady speed.

b. M-37 Truck

(1) The typical gear loadings in the high speed (Yuma area) portion of the field test are approximately equal to the gear loading of the high speed portion of the CRC L-37 test.

(2) The average axle speed in the high speed portion of the field test is very close to the axle speed in the high speed portion of the L-37 test.

(3) Although the dynamic gear loadings on the hills in the high torque (Death Valley) portion of field test varied considerably, the estimated time-average load would be reasonably close to the high torque loading of the L-37 test.

(4) The average axle speed on the hills in the high torque portion of the field test was nearly twice as great as the speed of the L-37 test.

(5) The shock torques produced in gear shifting vary considerably, depending upon the gears being shifted, the tractive effort at the time of the shift and the skill of the operator.

(6) Typical highway bumps or dips produced transient gear loads about 4 to 5 times the gear loading under level road load conditions at an equivalent speed.

c. M-211 Truck

(1) The largest shock loads produced by transmission shifting were approximately 70 per cent greater than the highest steady gear loads recorded on the steepest grades in Death Valley.

(2) The loading on the intermediate axle was higher than the loading on the rear axle under both shock loading and steady state operation.

IV. PHASE III - STUDY OF SLIDING VELOCITY OF HYPOID GEARS

A. Introduction

A primary purpose of this program has been to provide basic data on the lubrication environment of automotive hypoid gears. The work reported in Part III of this report covered the experimental investigation of dynamic loading of hypoid gears. It is apparent from a cursory analysis of the operation of hypoid gears that in addition to high unit loading on the gear teeth the severity of lubrication may also be traced to the high relative velocity between the mating ring and pinion gear teeth.

In order to investigate this portion of the lubrication environment of hypoid gears, a theoretical study was made of the sliding velocity of the same gears investigated in the previous study on dynamic loading of hypoid gears.

B. Object of the Study

The purpose of the work reported herein is to provide basic data on the sliding velocities of a variety of hypoid gears and to study the relationship of gear loading and sliding velocity upon the lubrication severity. In addition, the effect on gear sliding velocity of some gear design factors was studied to provide an indication of the influence on sliding velocity to be expected from current design trends or rear axle hypoid gears.

It is believed that this information will be valuable in providing a better understanding of the lubrication environment of hypoid gears and in estimating future lubrication requirements. These data would also be required for the future design of a small laboratory apparatus which may be developed to augment full scale axle testing of gear lubricants.

C. Gears Studied

In the previous work on dynamic loadings of hypoid gears, studies were conducted on gears in a number of gear lubricant tests. The same gear sets were used in this study to provide directly comparable data on both the loading and sliding of these gears.

The manufacturers of the four gear sets used in the dynamic torque studies were contacted and agreed to release the basic design data on the gear desired so that these computations could be carried out. The gear sets studied were those used in:

1. The CRC L-19 and the GM shock tests (4.11:1 ratio).
2. The CRC L-42 Test (3.92:1 ratio).

3. The CRC L-20, L-37 and in the Army M-37 truck (5.83:1 ratio).

4. The Army M-211 truck (6.17:1 ratio).

The kind assistance of the Chevrolet Division and GMC Truck Division of General Motors Corporation, the Dana Corporation and the Chrysler Corporation are gratefully acknowledged. Without their cooperation this work could not have been carried out. Thanks are also due to the Ford Motor Company and the Ethyl Corporation for assistance in providing the statistical information necessary for the study on design trends of hypoid gears.

D. Method of Computation

The sliding velocity of the above mentioned gears were computed using a method developed by the Gleason Works, Rochester, New York. This method contained in the Gleason Works pamphlet "Method for Designing Hypoid Gear Blanks" and "Sliding Velocity in Hypoid Gears" was made available to us for this program through the courtesy of the Gleason Works. The use of this method proved to be a great time-saver, in that independent development of a design and sliding velocity computation method was not necessary. The method as used by the Gleason Works was modified slightly in order to make it adaptable to an IBM 650 high speed digital computer.

The programming of the computation employed an interpretive system developed by Bell Laboratories referred to as the Bell L-1 Interpretive System. This system provides for programming in a simplified code language employing the floating decimal point. The interpretive system then translates the program instructions and data into the IBM 650 basic machine language. A computation time penalty results from the use of this system. But for a program of this size the extension of running time is of little importance compared to the convenience of programming. A block diagram of the computer program is shown in Appendix D.

E. Results and Discussion

The computation of the sliding in automotive hypoid gears has provided pertinent information in four general areas. It has given: (a) a more detailed understanding of the sliding relationships in these gears, (b) data on the sliding velocities of gears under standard test conditions, (c) data on those velocities under Yuma field tests, and (d) an understanding of the effect of design changes upon sliding velocity. In the discussion that follows each of these topics will be considered separately.

It should be noted that the calculations are of the sliding of the gears on the mean radius of the ring gear. The computation of sliding on other planes is greatly complicated by the intricate geometry involved. It is reasonable to suppose that the sliding velocity at the heel root of the gear tooth is greater than at the root on the mean diameter, but the values presented herein should provide an approximate average of the sliding which occurs on the teeth.

1. Sliding Relationships

The direction of slide on the mean radius varies across the profile of the teeth. At the bottom of the tooth contact pattern on the gear the angle between the horizontal and the direction of slide is a maximum and decrease to zero at the pitch line. In these calculations the direction of slide was computed across the tooth profile for all of the gear sets studied. As an example of the sliding of these gears a vector plot of sliding at various points on the mean radius of the Dodge truck axle is shown in Fig. 27. It will be noted that the direction of sliding in either the addendum or dedendum tends to become parallel to the pitch line as the pitch line is approached. Since the vertical component of sliding is approaching zero and the horizontal component is also decreasing as the

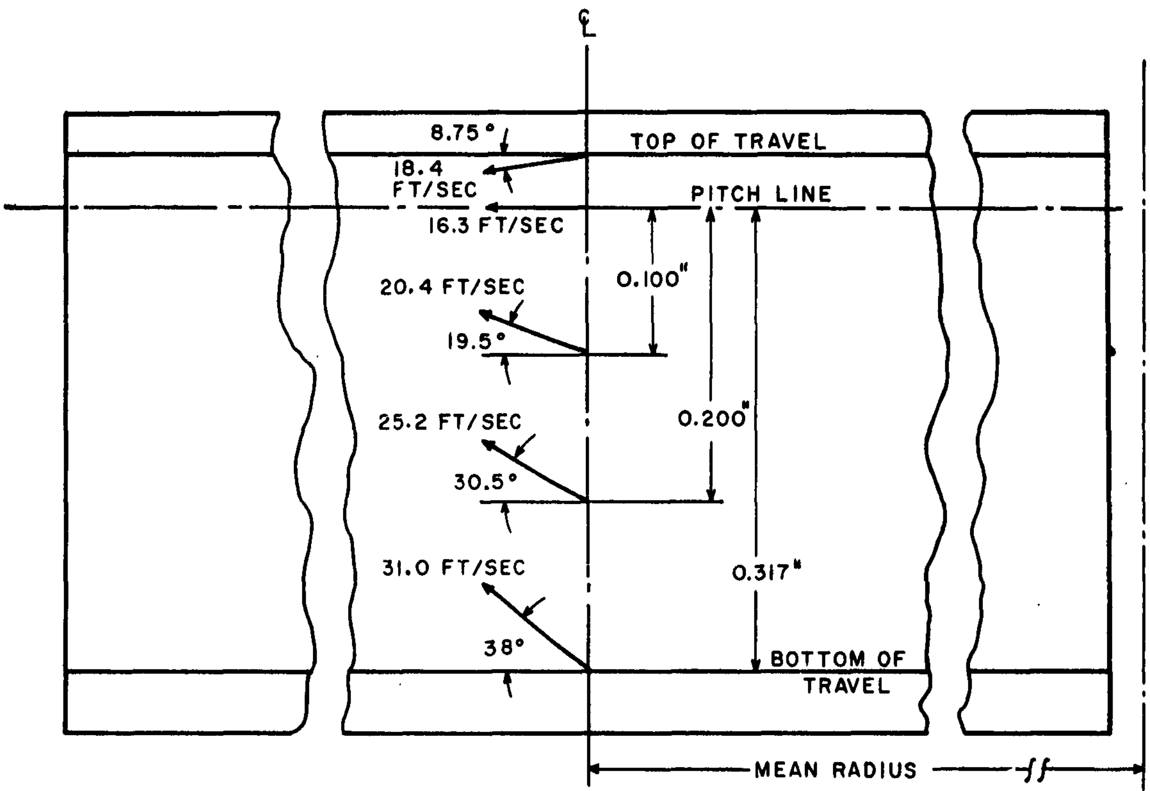


FIG. 27 - DIRECTION AND VELOCITY OF SLIDING ON M-37 TRUCK RING GEAR.

pitch line is approached, the vector sum of the relative sliding velocity is decreasing.

The direction of sliding of a hypoid gear is graphically shown by the pattern of "ridging" failure. Figure 28 is a photograph of "ridging" failure on the M-37 truck gear. The similarity in pattern between the ridging lines in the gear teeth and the direction of slide shown in Fig. 27 is self-evident. It is apparent that during ridging the metal is caused to flow in the direction of slide.

Figure 29 shows the angle of slide across the profile on the mean radius of the four gear sets studied. The maximum angle of slide was found on the M-211 truck gear with the other gears aligned in descending order of angle of slide as follows: M-37 truck, L-42 test gears, and the L-19 test gears. It may be noted that the angle of slide usually decreased (i.e., the longitudinal component of slide became relatively more effective than the vertical component) as the gear ratio decreased. However, L-42 and L-19 test gears fail to fall in order of their gear ratios. It is apparent that the slide angle is affected by a number of other factors in addition to the gear ratio. From an analysis of the geometry of the gears, it may be seen that some of the other design factors which influence relative sliding are mean gear radius, spiral angles, pressure angles and offset. The differences in mean gear radius, spiral angles and pressure angles between these two gears explain the greater angle of slide in the L-42 test gear than in the L-19.

As mentioned above, the velocity of slide increases on the mean radius for contact points farther from the pitch line of the gear. The sliding velocity of hypoid gears is directly proportional to the rotational speed of the gear. Therefore, in this analysis the term "specific sliding

FIG. 28 - RIDGING FAILURE ON M-37 TRUCK RING GEAR.

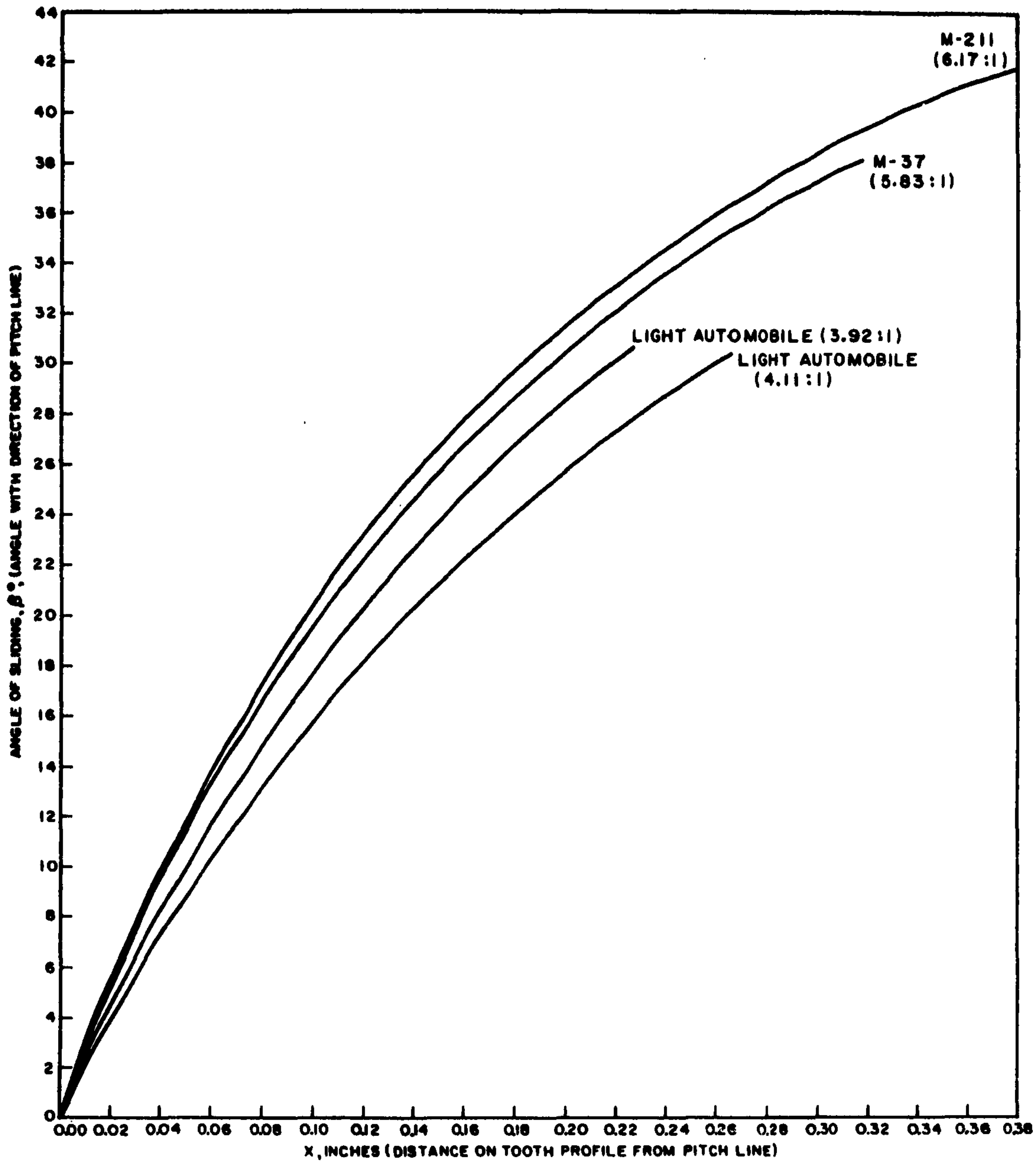


FIG. 29 - COMPARISON OF ANGLE OF SLIDE ACROSS PROFILE FOR FOUR AUTOMOTIVE GEARS.

velocity" is used to denote the rate of sliding of the contact points for the rotational speed of the ring gear. The units of this expression are feet per second per rpm of ring gear. The specific sliding velocities of each of the four gears studied are plotted in Fig. 30. The gears were aligned in order of decreasing specific sliding velocity as follows: M-211 truck gears, M-37 truck gears, L-19 test gears and L-42 test gears. It will be observed that as the gear ratio increased the specific sliding velocity increased. It would seem reasonable to expect greater sliding with a higher gear ratio.

2. Comparison of Sliding and Load in Full-Scale Lubricant Tests

Over a period of many years, a large amount of background information has been developed on extreme pressure lubricants which meet the military specification MIL-L-2105 and the new specification MIL-L-002105A. It is, therefore, of particular interest to determine the loading, sliding and temperature of the gear teeth under the conditions of the four full-scale tests required in the above specifications. In Section III summaries of the phase of the work on dynamic measurement of ring gear loading was presented. The work to date on the measurement of surface temperature of gear teeth will be discussed in a later section.

a. High Speed Gear Lubricant Procedures

Table 7 gives the unit loadings and maximum sliding velocities which are the important conditions of the CRC L-19 and CRC L-42 high speed shock tests. In order to make the data comparable between the various gear sets used the load data is given as lb/in-face of the ring gear. Figures 31 and 32 present comparisons of loading and sliding velocity for these two procedures.

Both test methods use a relatively lightly loaded run-in followed

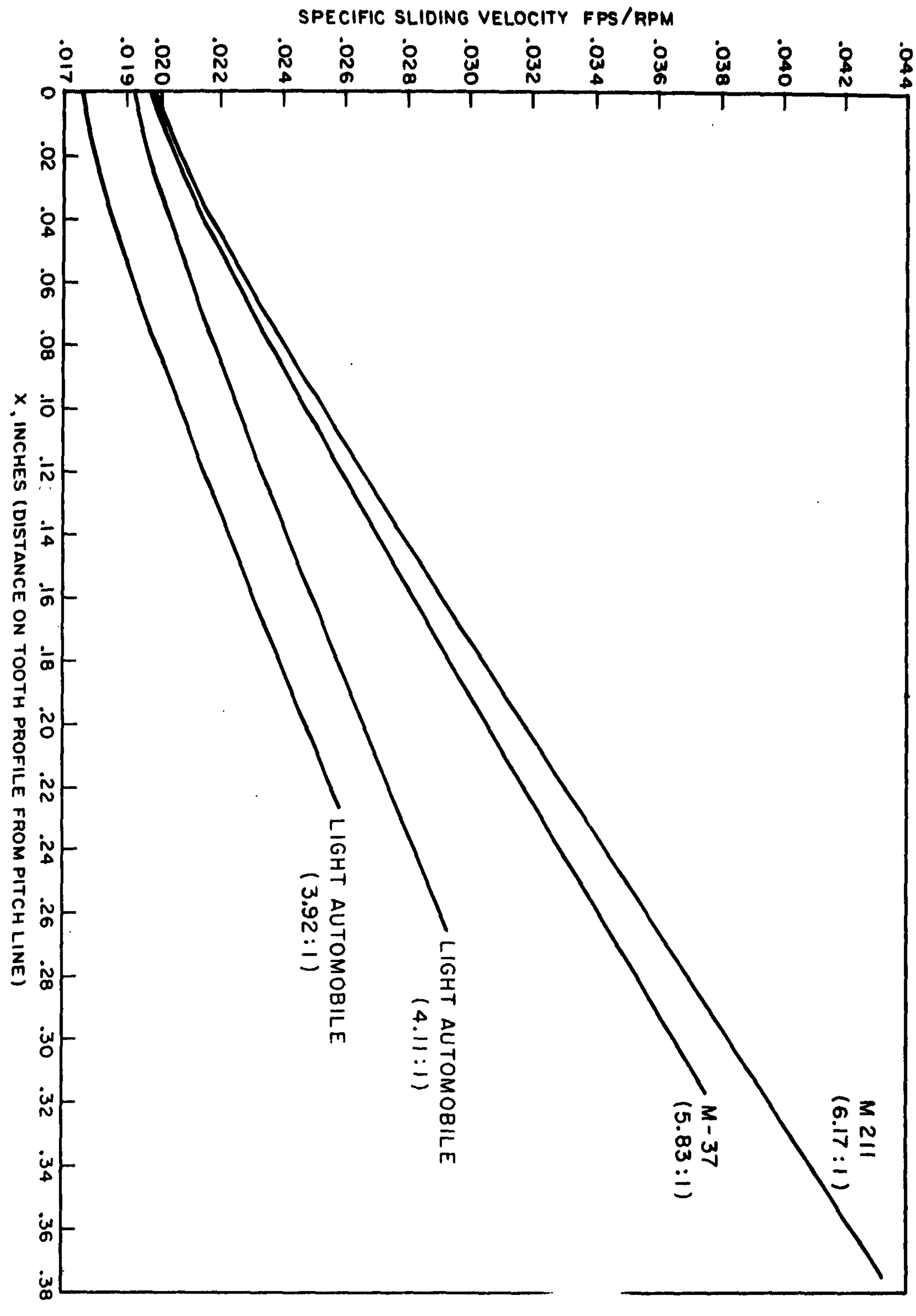


FIG. 30 – COMPARISON OF SPECIFIC SLIDING VELOCITY FOR FOUR AUTOMOTIVE GEARS.

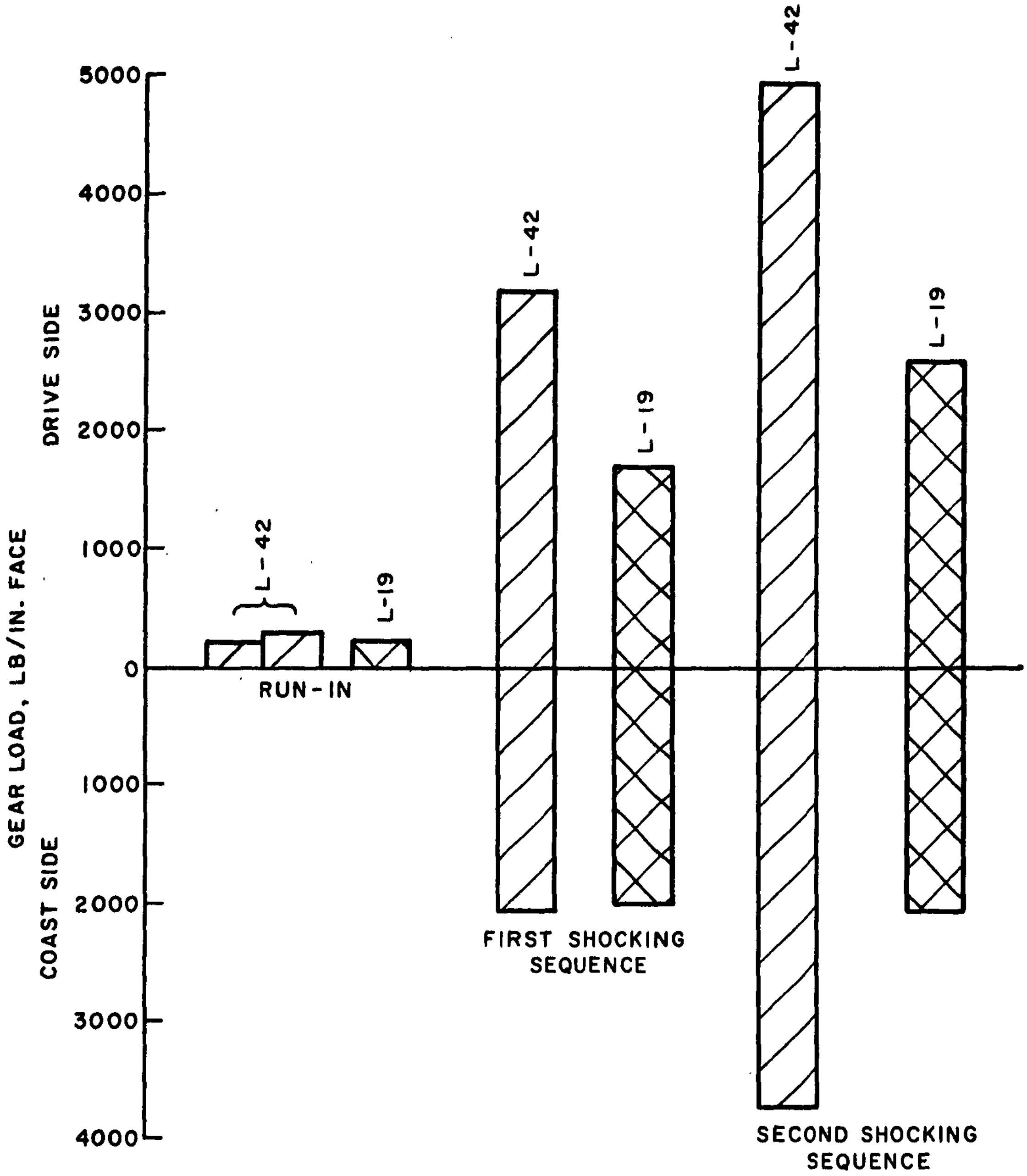


FIG. 31 - COMPARISON OF UNIT GEAR LOADING OF CRC L-19 AND L-42 HIGH SPEED GEAR LUBRICATION ATION PROCEDURES.

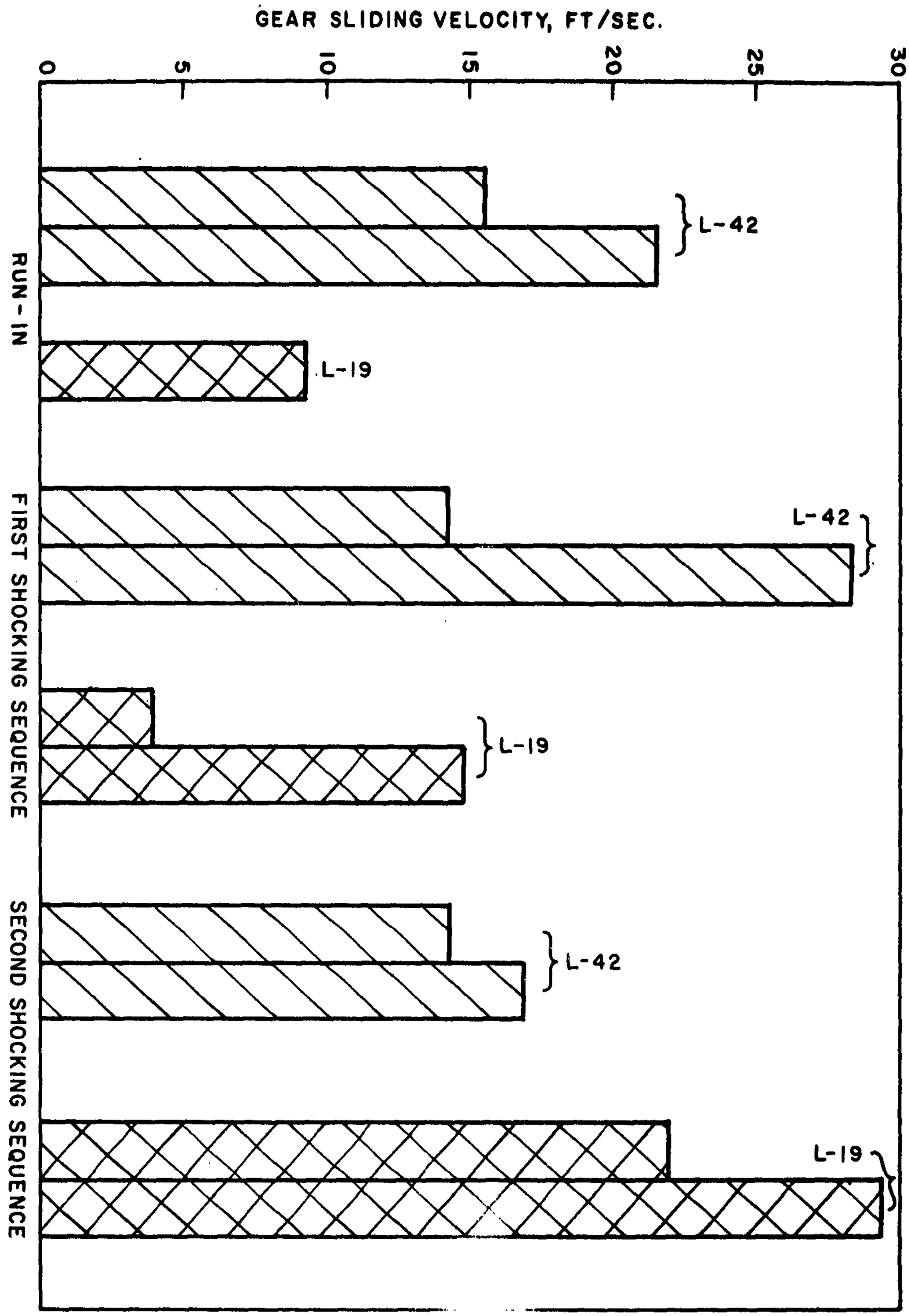


FIG. 32 – COMPARISON OF MAXIMUM GEAR SLIDING VELOCITY IN CRC L-19 AND L-42 HIGH SPEED GEAR LUBRICATION EVALUATION PROCEDURES,

by two shocking sequences. However, there are a number of differences apparent in the details of the test. The run-in of the L-19 test is at constant speed without contact on the coast side of the teeth except during the shifting of gears in the initial part of the run and the deceleration at the end of the run. The L-42 run-in incorporates two series of smooth, moderate accelerations and decelerations to provide a small amount of contact on the coast sides of the gears.

Table 7

COMPARISON OF HIGH SPEED SHOCK TEST GEAR CONDITIONS

Operating Condition		Axle Torque* lb/ft	Axle Speed rpm	Load* lb/in-face	Sliding Velocity ft/sec
CRC L-42					
Sequence #1	10 min	80d	600	202d	15.5
	20 min	104d	850	262d	21.9
Sequence #2					
Cycling between	min speed	1250d	550	3152d	14.2
	max speed	820c	1100	2068c	28.4
Sequence #4					
Cycling between	min speed	1950d	550	4917d	14.2
	max speed	1470c	650	3707c	16.8
CRC L-19					
Run-In (25 mph)		79d	312	211d	9.2
Cycling between	10 mph	625d	125	1671d	3.7
	40 mph	760c	500	2032c	14.7
Cycling between	60 mph	950d	750	2567d	22.0
	80 mph	770c	1000	2059c	29.4

* d - loading on drive side of gears; c - loading on coast side of gears.

From Table 7 and Fig. 31 it may be seen that the tooth loading in the run-in of both procedures is similar. However, by referring to Fig. 32 it is noted that the sliding velocity in the L-42 run is considerably higher in both portions than that in the L-19 run. During the development of the L-42 procedure, the speed of the run-in was found to be of great importance to the test repeatability. A number of variations in run-in were tried but the best repeatability and reproducibility were obtained when a run-in in the speed range of the test sequences was used. The reason for this effect is not obvious; however, it seems possible that the extreme pressure films produced under low load and low speed conditions may not be as resistant to abrasion as the films formed under higher speed conditions. Loeser[1], et al, reported a similar effect with EP films formed from zinc diakyl dithiophosphate on valve lifters and cams. They explained that it was "probable that a greater portion of the film formed under high speed and high load conditions has been converted to an inorganic compound which would be expected to be more resistant to abrasion." They further showed that an essentially stabilized film will be established under one set of load and speed conditions. When these conditions are suddenly increased, the first result is a sudden rapid decrease in film thickness followed by a recovery and a rapid buildup to thicker film than the original one. The work of Borsoff and Wagner[2] tends to substantiate these findings when applied to high speed spur gears.

If the film formed on hypoid gears under a low speed run-in is less persistent and more susceptible to attrition due to abrasion than the

---

[1] Loeser, E. H., Wiquist, R. C. and Twiss, S. B., "Radioactive Tracers in Extreme Pressure Film," paper presented at Nuclear Engineering and Science Conference of SAE, March 17-21, 1958.

[2] Borsoff, V. N. and Wagner, C. D., "Studies in Formation and Behavior of an Extreme Pressure Film," presented at ASLE Lubrication Conference, October 10-12, 1955.

film formed by a higher speed run-in, the former film would deteriorate more rapidly under the first shock loads of the first sequence. Being less persistent these films may be attrited erratically causing a wide spread in the gear scoring results of the test. At present no directly applicable data is known which would either substantiate or disprove this supposition. Further investigation of the effect of sliding speed on the surface temperature and the resultant effect on film formation should be carried out at an extension of the work on this project.

The second part of the L-19 test is a low speed cycling sequence. It consists of a drive side shock of about 1670 lbs/in.-face at a sliding velocity of 3.7 ft/sec. and an average coast side shock of approximately 2030 lbs/in.-face at a sliding velocity of 14.7 ft/sec. The second sequence of the L-42 test is a similar cycling run but produces average drive side shocks of about 3150 lbs/in.-face at 14.2 ft/sec. and an approximate average coast side load of 2070 lbs/in.-face at 28.4 ft/sec. Thus, the sliding velocities of Sequence 2 of the L-42 procedure may be seen to be at least twice as great as those in the comparable portion of the L-19 procedure. The loads on the drive side in this sequence of the L-42 are nearly twice as great as those in the L-19 but roughly equal on the coast side of both procedures. The higher sliding velocities and higher drive-side loads of the L-42 procedure would indicate a more severe lubricant requirement for this portion of the L-42 test as compared to that same portion of the L-19 test. This is confirmed by observations made after this portion of the test in both the L-19 and L-42 tests. Surface distress was seldom noted in the L-19 test after this portion of the test even though the lubricant quality might be less than half as great as that required to pass the L-42 test. There is usually at least a trace of surface disturbance visible on the ring gear after this portion of the L-42 test even with a passing lubricant under this procedure.

The final portion of the L-19 test is a relatively high speed cycling sequence. This cycle produces an average drive side shock of approximately 2570 lbs/in.-face at a sliding velocity of 22.0 ft/sec. and an average coast side shock of about 2060 lbs/in.-face at a sliding velocity of 29.4 ft/sec. The comparable portion of the L-42 test produces an average drive side shock of about 4917 lbs/in.-face at a sliding velocity of 14.2 ft/sec. and an approximate average coast side shock of 3707 lbs/in.-face at a sliding velocity of 16.8 ft/sec. The sliding velocity of this portion of the L-42 test is considerably less than that in the comparable portion of the L-19 test. On the other hand, the loadings of the L-42 tests are nearly twice those in the L-19 test on both the drive and coast sides of the gears.

b. High Torque Gear Lubricant Procedures

A comparison of the load and sliding velocities of the CRC L-20 and L-37 high torque low speed tests is shown in Fig. 33. Table 8 tabulates the loads and sliding velocities of the gear teeth under the two portions of each test.

Table 8

COMPARISONS OF HIGH TORQUE STEADY SPEED TEST GEAR CONDITIONS

Operating Condition	Axle Torque lb/ft	Axle Speed rpm	Load * lb/in.-face	Sliding Velocity ft/sec.
L-37				
100-min high speed sequence	788	440	1501	16.6
24-hr low speed sequence	3483	80	6633	3.0
L-20				
Run-In	500	62	952	2.3
30-hr low speed run	2693	62	5129	2.3

* All test loading is on drive side of gears.

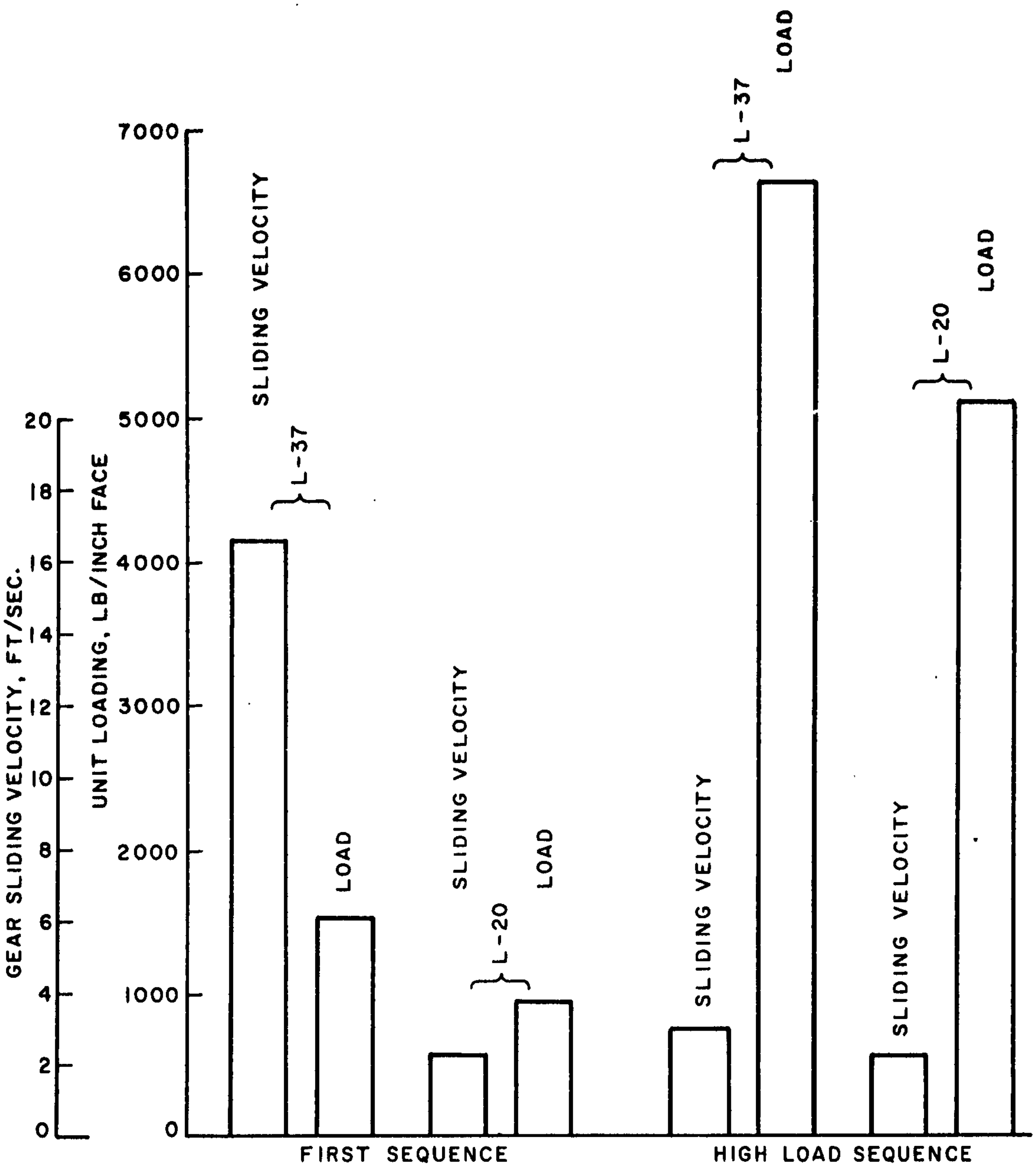


FIG. 33 - COMPARISON OF MAXIMUM GEAR SLIDING VELOCITY AND LOAD IN CRC L-20 AND L-37 EVALUATION PROCEDURES.

The L-20 test commences with a run-in which is continued until the bulk oil temperature in the differential housing reaches 140°F. This temperature rise usually requires about 20 minutes. During that time a steady load of about 950 lbs/in.-face is maintained at a sliding velocity of about 2.3 ft/sec. In the L-37 test there is no run-in, per se, but the first portion of the test is equivalent to the operation of a heavily loaded truck and trailer travelling at 50 mph over gently rolling country (see Section III of this report). The load applied to the gear is about 1500 lbs/in.-face at a sliding velocity of about 16.6 ft/sec. It is obvious that the initial conditions of the L-37 test are several times more severe than the run-in of the L-20 test. In our laboratory experience with the L-37 test, we have noted ridging failure of the gears after the 100-minute run under these conditions on numerous occasions. Some sort of surface disturbance would be expected after this portion of the L-37 test on lubricants providing a marginal passing performance in the L-20 test. As observed in the development of the L-42 test the design of the initial portion of the test is an important factor in determining the severity level and repeatability of the full-scale gear lubricant test.

The second sequence of the L-20 test is run for 30 hours at a load of about 5130 lbs/in.-face and a sliding velocity of 2.3 ft/sec. The second portion of the L-37 test is run for 24 hours at a load of approximately 6630 lbs/in.-face at a sliding velocity of 3.0 ft/sec. The L-37 loading during the second sequence may be seen to be more than 1-1/4 times as great as the L-20 loading and the sliding 1-1/3 times as great as the L-20.

3. Sliding Velocity of Army Truck Gears in Yuma Field Tests

The results of the study of dynamic loading of the driving axles of two Army trucks which were tested at the Yuma Test Station in Arizona are presented in Section III above. By using the sliding velocity computation methods discussed above the velocity of sliding of the contact areas were obtained for the axle drive gears of both the M-37 and the M-211 trucks.

Figure 34 presents a comparison of the sliding velocities of the gears from both trucks under the same operating conditions as described in Section III above. The sliding velocities of both gears under these and several other operating conditions are shown in Tables 9 and 10. It may be observed that in nearly all steady-load conditions in which a reasonably heavy load is applied to the gears (all hill climbing conditions), the sliding velocity of the M-37 truck gears is greater than that of the M-211 gears. As has been noted previously, the loading on the M-37 gears is higher under comparable test conditions than the loading on the M-211 gears. This, coupled with the higher sliding velocities for comparable portions of the test run for the M-37 gears, would tend to contribute to the greater lubrication failure rate observed on the M-37 truck gears under the Yuma field test conditions.

Tables 11 and 12 present the unit loading and sliding velocity data on the M-37 and M-211 truck gears under shock loadings which resulted from gear shifting. The highest gear loadings recorded during the test occurred as shocks during gear shifting. The greatest shock loading in the M-211 truck occurred on down-shifting from high-second to low-fourth while travelling downhill. Unit loadings of about 8084 lbs/in.-face and 7218 lbs/in.-face were measured on the intermediate and rear axles, respectively. The gear sliding velocity when this shock occurred was approximately 2.7 ft/sec. The maximum shock load measured for the M-37 truck also occurred during down-shifting. However, the greatest shock was recorded while travelling uphill rather than downhill as in the M-211 truck. Therefore, the greatest

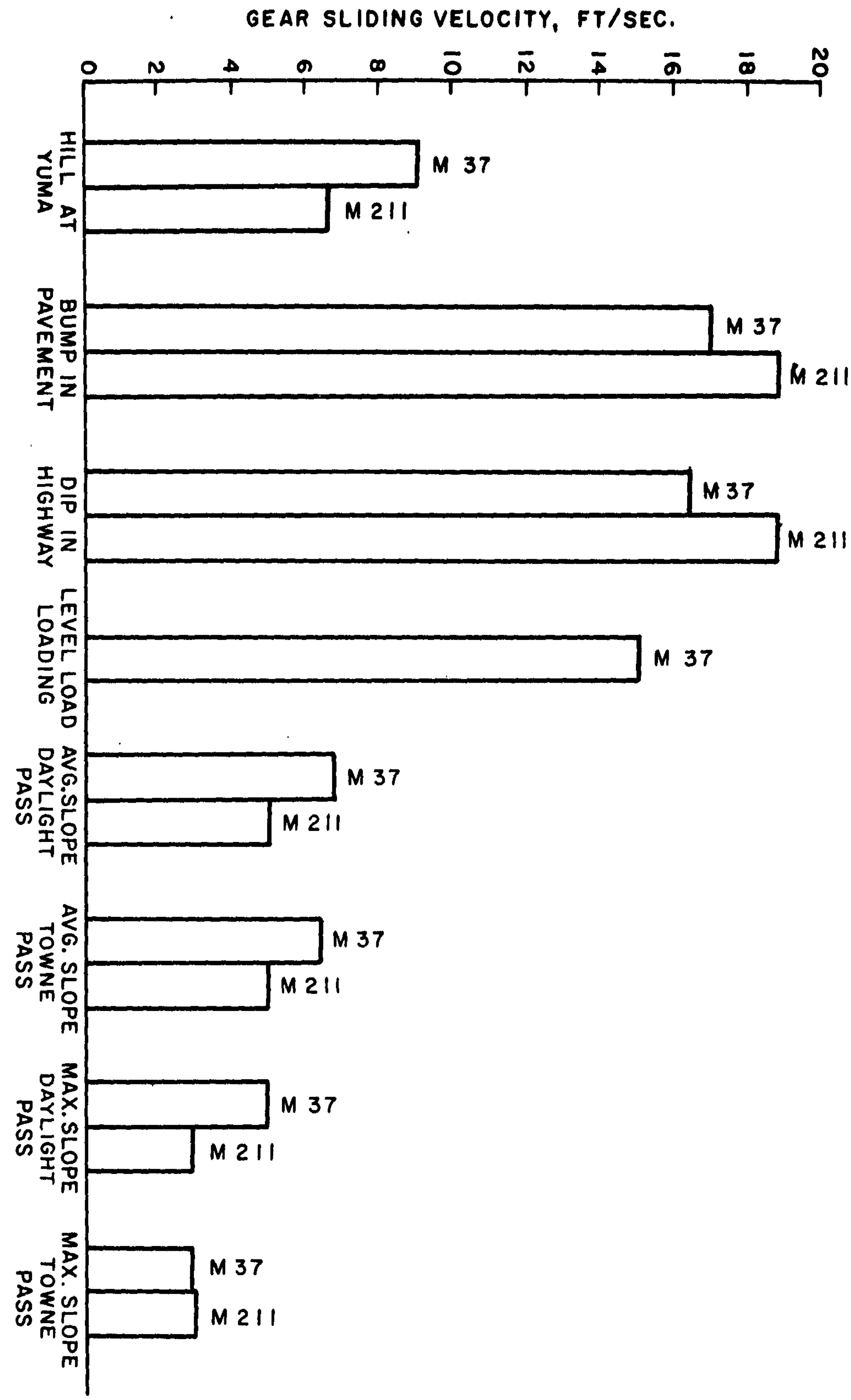


FIG. 34 – COMPARISON OF SLIDING VELOCITY OF M 37 AND M 211 TRUCK GEARS IN YUMA FIELD TESTS.

Table 9

AVERAGE GEAR LOADS AND SLIDING VELOCITIES ON M-37 VEHICLE YUMA HIGHWAY PORTION TEST COURSE

	Gear[1] Load lb/in.-face	Sliding Velocity, ft/sec.	Gear Range
Maximum Steady Load on Hill at North End of Course			
Uphill	6447d	9.130	H-3
Downhill	171c	17.689	H-4
Typical Bumps in Pavement	844c	16.966	H-4
	3588d	-	-
Typical Dip in Pavement	3009d	16.358	H-4
Steady Speed Level Road Load	177d	5.896	H-4
	491d	11.450	H-4
	632d	15.293	H-4

Table 10

AVERAGE GEAR LOADS AND SLIDING VELOCITIES ON M-211 VEHICLE YUMA HIGHWAY PORTION TEST COURSE

	Intermed.[1] Gear Load lb/in.-face	Rear Axle[1] Gear Load lb/in.-face	Sliding Velocity, ft/sec.	Gear Range
Maximum Steady Load on Hill at North End of Course[2]				
Uphill	2003d	1517d	6.643	H-2
Downhill	143c	465c	18.506	H-4
Typical Bumps in Pavement[2]	1142d	609d	18.911	H-4
Typical Dip in Pavement[2]	808d	258d	18.774	H-4

[1] d - loading on drive side of gears; c - loading on coast side of gears.

[2] Towing trailer of 5500 lb. gross weight.

Table 11

AVERAGE GEAR SHOCK LOADS AND SLIDING VELOCITIES IN M-37 TRUCK

DUE TO GEAR SHIFTING

Gear Shifts	Slope of Road	Gear Load[1] lb/in.-face	Sliding Velocity ft/sec.
Low Gear Range			
From low 3 to low 4	Uphill	1209c	8.483
		6494d	-
	Downhill	2483c	7.342
From low 4 to low 3	Uphill	3989c	7.418
		10494d	
	Downhill	6618d	7.418
		1923c	

Table 12

AVERAGE GEAR SHOCK LOADS AND SLIDING VELOCITIES IN M-211 TRUCK

DUE TO GEAR SHIFTING

Gear Shifts[2]	Slope of Road	Intermed. Gear Load[1] lb/in.-face	Rear Axle Gear Load[1] lb/in.-face	Max. Sliding Velocity ft/sec.
High Gear Range				
From high 1 to high 2	Level	4113d	3255d	5.375
From high 2 to high 3	Level	4378d	3505d	8.096
	Uphill	4042d	3259d	8.173
From high 4 to high 3	Level	4163c	3047c	3.810
From high 3 to high 2	Uphill	3223c	3180c	6.025
		4268d	3381d	-
From high 2 to high 1	Level	2332c	1645c	2.882
Low Gear Range				
From low 3 to low 4	Uphill	5384d	4983d	3.598
From low 4 to low 3	Uphill	6908d	6487d	3.187
Shifting Between High & Low Range				
From low 4 to high 2	Uphill	3291d	2325d	4.879
From high 2 to low 4	Downhill	8084c	7218c	2.699

[1] d - loading on drive side of gears; c - loading on coast side of gears.

[2] Towing trailer of 5500 lb. gross weight.

loading was on the drive side of the teeth. With this gear shift a 3989 lbs/in.-face load was first applied to the coast side of the teeth and then the loading reversed to apply a 10,494 lbs/in.-face load on the drive side of the teeth. At the time of this load application, the sliding velocity of the gear was approximately 7.4 ft/sec. This drive side load of approximately 10,500 lbs/in.-face is second in magnitude to the loading of the laboratory GM shock test (about 14,000 lbs/in.-face) of any loads measured in this program. It is quite apparent that this shock loading and sliding velocity of the M-37 truck gear presents a greater lubrication problem than was present in the M-211 truck under its maximum shock loading conditions.

4. Present Gear Design Trends and Their Effects on Sliding

As mentioned above the sliding velocity is affected by several gear design factors, including mean gear radius, spiral angles, pressure angles, gear ratio and pinion offset. It is apparent that in a practical gear design these factors are not independent variables but to a great extent the selection of a value for one of these parameters is mutually dependent upon the values selected for the others. The gear designer has a certain amount of freedom of selection but only within fairly narrow limits imposed by the over-all design concept of the gear assembly.

Since the design of an automotive hypoid gear is so highly dependent upon the rest of the engine-transmission system and is strongly influenced by the space available to the axle and drive shaft due to body styling, it is believed that the two most nearly independent design factors are the pinion offset and the gear ratio. A study of the post-war trends in automotive hypoid gear design reveals definite trends in the industry in both pinion offset and gear ratio.

A comparison of the pinion offset of 1958 passenger cars with that of 1946 automobiles is shown in Fig. 35. It may be seen that whereas the lowest offset used has not changed (1.375 in.) the greatest offset used has increased from 1.75 in. to 2.375 in the twelve-year period. An average value of offset was computed, weighted on the basis of the automobile population. It may be observed from Fig. 35 that this weighted average pinion offset has increased from 1.54 in. in 1946 to 1.77 in. in 1958. During this same period there has been very little change in the gear diameter.

A study of the trend in gear ratios in passenger automobiles is shown in Fig. 36. The maximum and minimum gear ratios for each model year are plotted along with the weighted average gear ratio. It will be observed that the trend of the average gear ratio has been consistently downward. This trend has progressed at an accelerated rate since 1952. At about this time the vehicle designers began to use a significantly lower ratio for cars equipped with automatic transmission than for those equipped with standard transmission. The torque multiplying functions of automatic transmissions have resulted in a desire for a lower rear axle ratio. The predominence of automatic transmissions (estimated to be in nearly 80 per cent of the 1958 automobiles) has brought the average gear ratio much closer to the minimum than was true in 1952 or 1953. The gear with the maximum ratio for each year of this study has been on an automobile equipped with overdrive. The gradual decrease in the quantity of automobiles equipped with overdrive (estimated to be about 3.5 per cent in 1958) has resulted in a smaller influence of these higher ratio gears on the average.

With the present trends in both pinion offset and gear ratio, it was considered important to determine the effect of changes in these parameters on the sliding velocity of hypoid gears. Computer runs were made

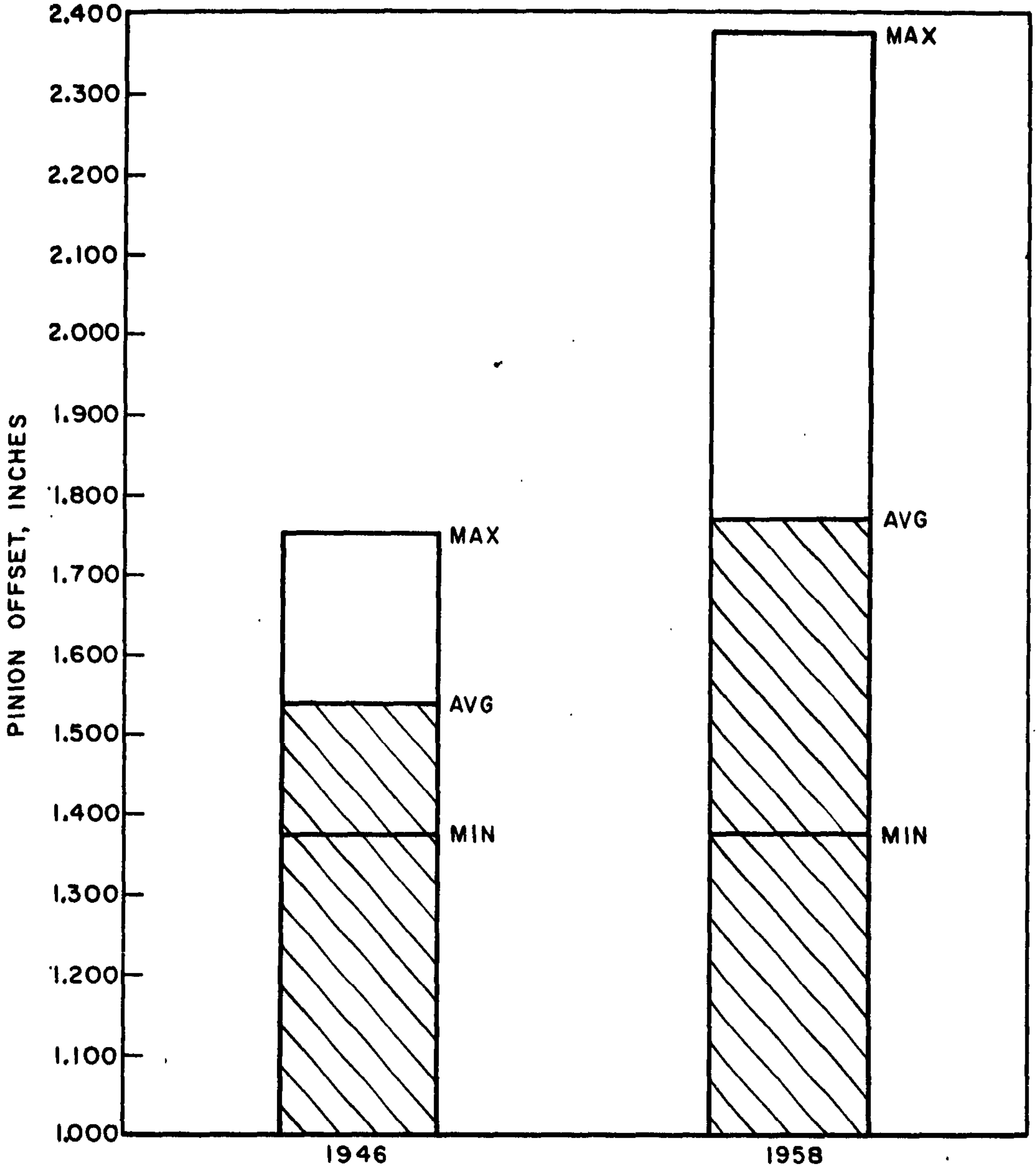


FIG. 35 - PINION OFFSET OF AMERICAN PASSENGER AUTOMOBILE HYPOID GEARS.

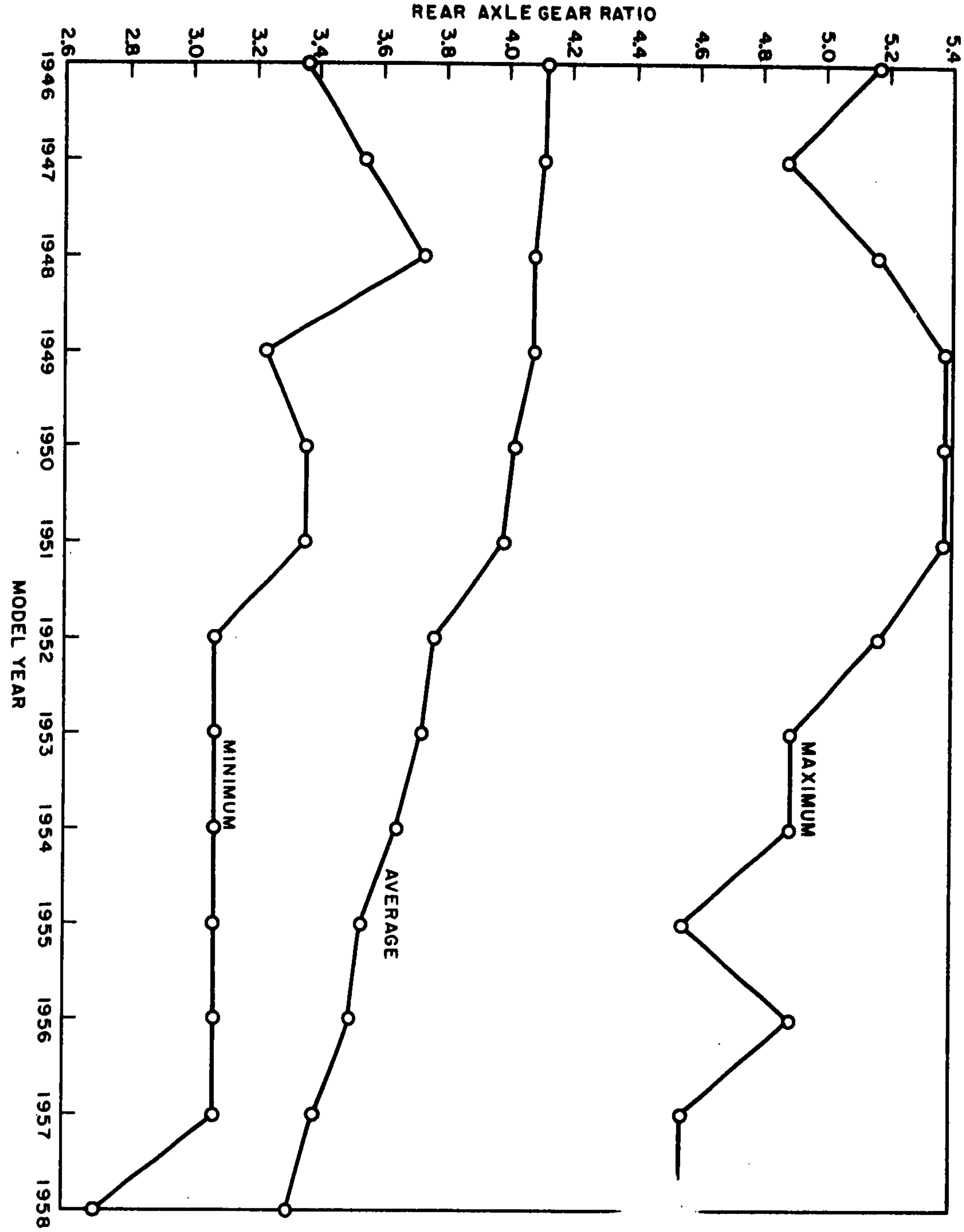


FIG. 36– TREND IN GEAR RATIO OF PASSENGER CARS.

of the gears from the L-42 test which were selected as fairly representative of current automotive practice to determine the effect of these two design factors. With all other factors kept constant, a series of runs were made to determine the sliding velocity of this gear set by employing a series of pinion offsets ranging from 1.25 in. to 2.50 in. As will be noted from Fig. 35 this range covers all values of offset used from 1946 to the present.

The relationship of pinion offset to sliding velocity is plotted in Fig. 37. It may be observed that the maximum specific sliding velocity varies directly with pinion offset. An increase of offset in this gear of from 1.25 in. to 2.50 in. resulted in a change in specific sliding velocity from 0.0233 ft/sec/rpm. to 0.0359 ft/sec/rpm.; doubling the offset resulted in approximately 54 per cent greater sliding velocity in this gear. Referring to Figs. 35 and 37, it may be seen that a change equal to the change in average pinion offset from 1946 to 1958 would cause an increase of approximately 8.9 per cent in sliding velocity in this hypothetical case.

It should be kept in mind that these calculations would not represent a practical gear since any sizable change in one of the major design factors would doubtless be accompanied by changes in other design factors which would also tend to affect the sliding velocity. These calculations were performance to provide basic data on the trends in sliding velocity to be expected by further design changes in hypoid gears.

It is apparent from Fig. 35 that, barring a radical departure in automotive styling and drive systems, the trend will be toward higher offset gears. It is equally apparent that this design trend will tend to increase the sliding velocity of automotive hypoid gears and thereby increase the lubrication requirement. An accurate determination of the relative

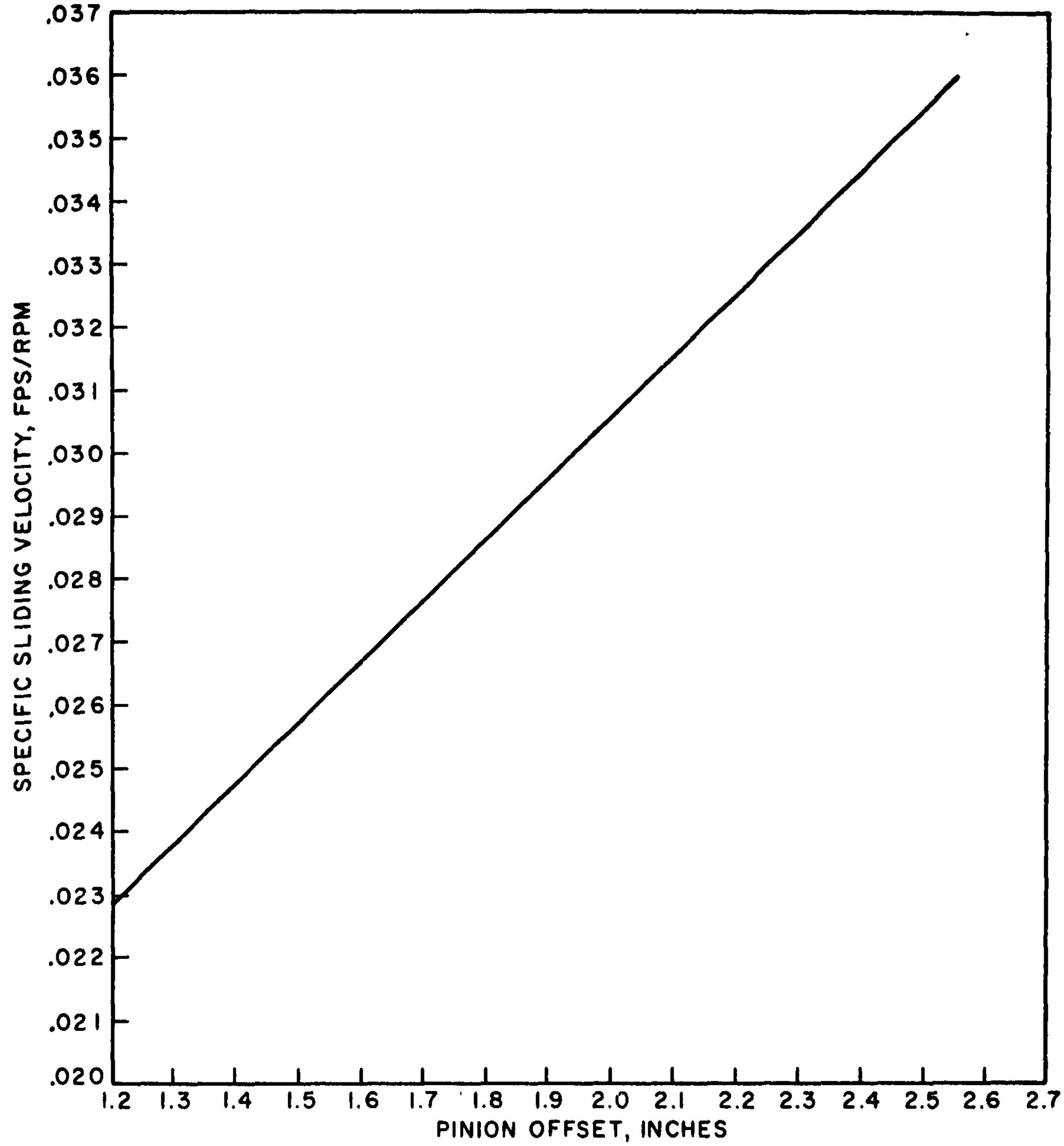


FIG. 37 - EFFECT OF PINION OFFSET ON SLIDING VELOCITY.
(HYPOTHETICAL GEAR SET)

importance of sliding velocity in gear loading is not available at this time and remains an important deficiency as well as a most difficult and complicated one to satisfy by either analytical or experimental means. Therefore, no numerical expression of the increase in lubrication requirement which can be expected as a result of an increase in pinion offset can currently be made.

A study of the effect of gear ratio on sliding velocity was conducted similarly to the study of pinion offset. With all other design factors of the L-42 test gears kept constant, the gear ratio was changed and the sliding velocity computed. Gear ratio values ranging from 2.60 to 4.09 were used. This range covers the average gear ratios from about 1946 to the present and also includes the smallest ratio presently in use. The results of these computations are plotted in Fig. 38. The specific sliding velocity was also found to vary directly with ratio. As the gear ratio was increased from 2.6 to 4.0 the maximum specific sliding velocity increased from 0.0221 ft/sec/rpm. to 0.0262 ft/sec/rpm. which is about 18.6 per cent. On this hypothetical gear a change in gear ratio equal to the change in average gear ratio from 1946 to 1958 would decrease the specific sliding velocity about 8.1 per cent. As mentioned above, a change equal to the change in average pinion offset from 1946 to 1958 would cause an increase of approximately 8.9 per cent in the specific sliding velocity of this gear set. Therefore, based upon the hypothetical case used herein the net effect of the changes in average pinion offset and average gear ratio on gear sliding velocity has been very small. Of course, this would not be true of individual gear sets which had been changed predominantly in either gear ratio or offset.

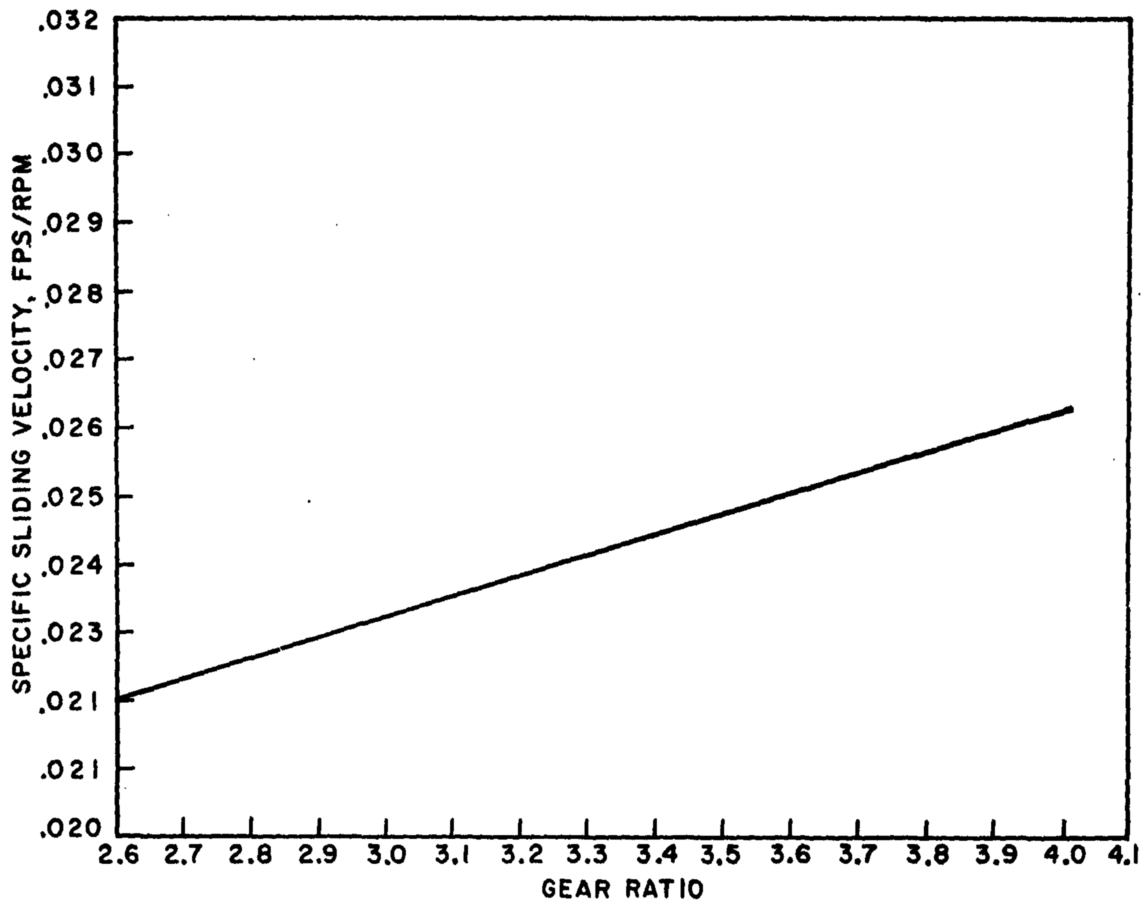


FIG. 38 - EFFECT OF GEAR RATIO ON SLIDING VELOCITY.
(HYPOTHETICAL GEAR SET)

F. Conclusions

1. General

a. The velocity of relative sliding for hypoid gears is a minimum at the pitch line and increases for contact points farther from the pitch line.

b. The direction of slide at the pitch line is parallel to the pitch line and the angle between the direction of slide and the direction of the pitch line increases with the distance from the pitch line.

c. The sliding velocity and angle of slide are affected by the gear ratio, pinion offset, mean gear radius, spiral angles and pressure angles.

d. Present gear designs are tending toward greater pinion offset. The weighted average offset in passenger cars from 1946 to 1958 has increased from about 1.54 in. to 1.77 in. In the same period the maximum offset used has increased from 1.75 in. to 2.375 in.

e. An increase of pinion offset increases the sliding velocity. In a hypothetical case, an increase in pinion offset equal to the average change in passenger cars since World War II has resulted in an approximately 8.9 per cent increase in sliding velocity.

f. Present gear designs are tending toward lower gear ratios. The weighted average gear ratio of passenger automobiles has decreased from about 4.11:1 in 1946 to about 3.31:1 in 1958.

g. A decrease in gear ratio results in a decrease in sliding velocity. In the hypothetical case used in this study, a decrease in gear ratio equal to the average decrease in ratio in post-war gears resulted in a decrease of about 8.1 per cent in sliding velocity. Therefore, the net effect on sliding velocity of the average changes in pinion offset and gear ratio since 1946 has been small.

2. Gear Sliding in Laboratory Evaluation Techniques

a. The gear sliding velocities in the CRC L-42 high speed procedure are higher than those of the CRC L-19 in all comparable portions of the test except in the final sequence. During the run-in and the first sequence, the L-42 sliding velocities are nearly always twice those of the L-19 test.

b. The sliding velocities in the CRC L-37 high torque procedure are higher than those of the CRC L-20 procedure in both portions of the tests. In the initial portion of the tests the sliding velocity of the L-37 is about seven times as great as the L-20. In the high torque portions the L-37 sliding velocity is about 1.3 times as great as that of the L-20.

3. Gear Sliding in the Yuma Tests

a. Both the sliding velocities and the unit loadings of the M-37 truck were nearly always greater than those of the M-211 truck both under steady loading and shocking from gear shifting.

b. The average sliding velocity of the M-37 truck gears in the Yuma portion of the test course was nearly equal to the sliding velocity of the high speed portion of the L-37 test. The average sliding velocity in the Death Valley portion of the test course was approximately twice the sliding velocity of the high torque portion of the L-37 test.

V. PHASE IV - STUDY OF UNIT SURFACE LOADING OF HYPOID GEARS

A. Introduction

One of the primary purposes of the work in this project has been to determine the reaction of extreme pressure lubricants under the environment imposed upon them in the rear axle gear case of automobiles and trucks. Whereas considerable excellent research has been applied to the study of lubrication of spur gears especially in the application of high speed spur gears for aircraft use, little has been done with the fundamentals of lubrication with automotive hypoid gears. If the loading, sliding and surface temperature of automotive hypoid gears can be determined in terms of fundamental measurements, it should then be possible to use the fundamental knowledge obtained from spur gear lubrication research in understanding the mechanism of lubrication from the hypoid gear. In order to accomplish this it is necessary to get the loading, sliding and temperature factors of the hypoid gear in basic terms of measurement which can be related to those already determined for the spur gears. For this reason, it has been considered important to reduce the gear loading information discussed in Section III above into terms of unit loading between two mating gear teeth rather than lb-ft. of torque or lbs/in.-face of the gear teeth. For this reason work was undertaken under this project to determine the unit loading of hypoid gear teeth from the experimentally determined torque measurements reported above.

Whereas progress has been made on this portion of the project, time and funds did not permit the completion of this work. The following section will present information on the progress to date in this phase of the work.

B. Methods of Study

Two independent methods are being used to study the surface loading of hypoid gears. The first, a theoretical method, applies the theory of elasticity to the geometry of the hypoid gear in order to calculate the Hertzian stress on the gear teeth. As a check on the theoretical approach, the determination of contact area is being pursued by means of experimentally measuring the electric resistance of the contact of two gear teeth.

C. Results and Discussion

1. Theoretical Study

The approach used in this study is based upon some initial work first published by Coleman.[3] Whereas his work was aimed at providing the formulas for the determination of root stress in the teeth and not directly

---

[3] Coleman, Wells, "An Improved Method for Estimating the Fatigue Life of Bevel Gears and Hypoid Gears", paper presented at SAE Summer Meeting, June 3-8, 1951.

for determining unit surface loading, the initial portion of his calculations are useful for the purpose of this study.

The following assumptions are being made for the derivation:

a. The theoretical contact of a hypoid gear tooth is a line (assuming no deformation of the tooth).

b. The actual contact line does not stretch across the entire tooth surface because of the intentional mis-match of the gears in both longitudinal and lateral directions. This results in the swept area approximating an ellipse, the axes of which are the face width of the gear tooth (F) and the height of the tooth minus the relief at the root (Z) (see Fig. 39).

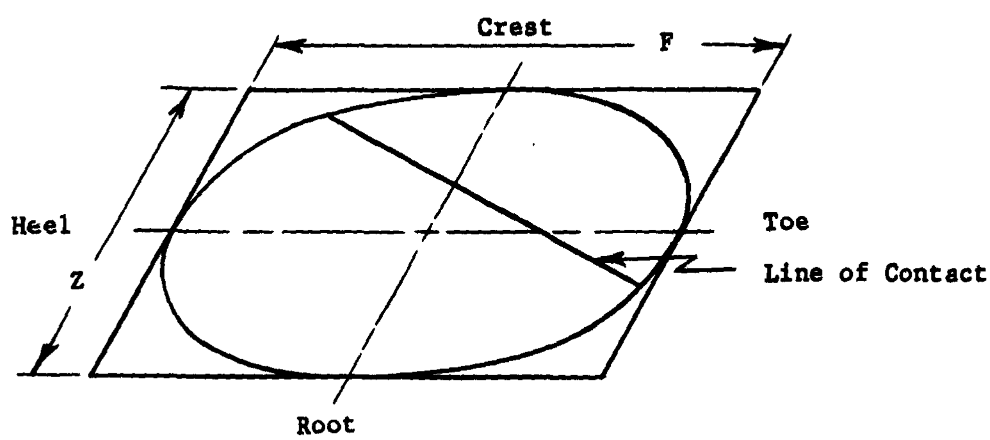


Fig. 39 THEORETICAL LINE OF CONTACT AND SWEPT AREA OF CONTACT FOR HYPOID GEAR (assuming no deflection)

c. Since the gear material is elastic and does deform under load, the contact becomes an area rather than a line because of the bi-directional mis-match of the gears, the contact area is assumed to be an ellipse as shown in Fig. 40.

d. Since the unit loading is related to the surface contours of the gear areas in contact, the loading is distributed as shown in Fig. 40.

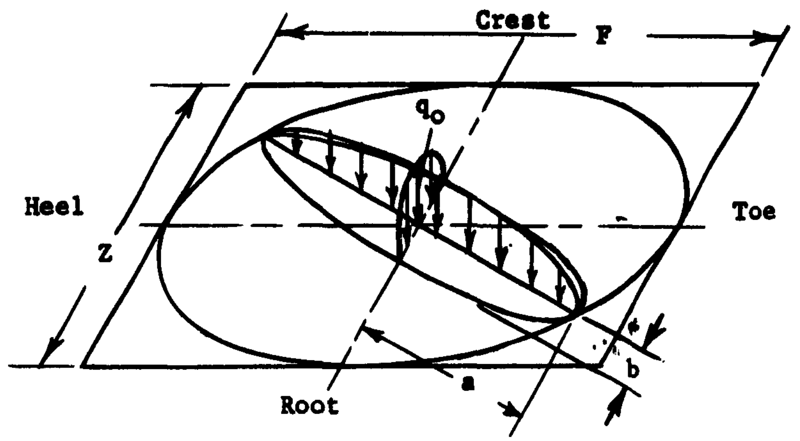


Fig. 40 THEORETICAL AREA OF CONTACT FOR HYPOID GEARS
(assuming elastic material)

e. The total load on the gear teeth would be:

$$P = \iint q \, d\, A$$

This equation may be solved to give

$$P = 2/3 \pi a\, b\, q_o \quad \text{or} \quad q_o = 3/2 \; \frac{P}{\pi\, a\, b}$$

f. Using the relationships or an ellipse, the total load may be expressed by the solution of equation above in terms of q_o, the maximum unit stress in the contact ellipse and the axes of the contact ellipse, b and a. Since the length of the chord (2a) of the swept-area ellipse may be calculated, the other axes of the contact ellipse (b) remains to be evaluated.

The length of the transverse axes of the contact ellipse is a function of the relative curvatures of the two surfaces in contact. In other words, the length of the contact axes (b) is a function of the shape of the tooth surface of the ring gear and pinion at the point of contact. In order to determine the relative curvatures of the mating teeth, further information is required beyond that presently available to our laboratory. It is believed that this information can be derived from basic design data available at the Gleason Works. It is intended that in the continuation of this contract this data will be obtained from the Gleason Works and the determination of the relative contours of the mating surfaces will be accomplished.

As noted above, this analysis assumes that the contact area at any point in time is in the form of an ellipse. From the spiral bevel gear etching test conducted by the General Motors Research Laboratories[4], it is apparent that the contact pattern is not an ellipse but as shown in Fig. 41. It may be seen that the center of the contact area is higher on the gear tooth than had been originally assumed. This face would appreciably affect the root stresses of the gear teeth but is not expected to materially change the determination of the unit loading or the total area in contact. Because of the extra complications which would be introduced by an attempt to use the actual contact pattern in these calculations, this work will be conducted on the assumption that the contact area is an ellipse but accounting for the more accurate position of the maximum unit stresses.

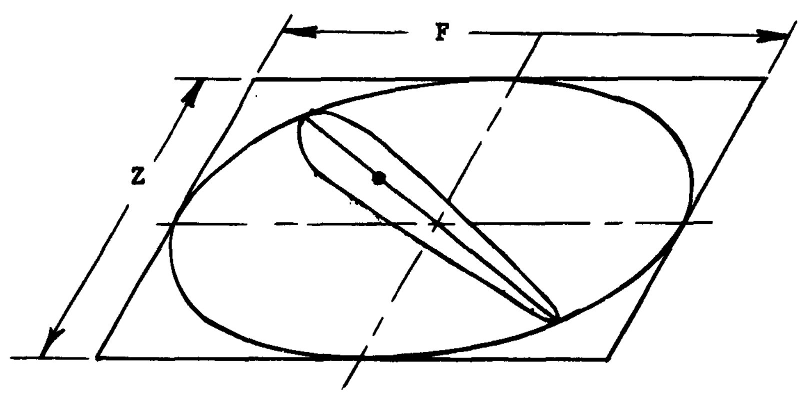


Fig. 41 ACTUAL CONTACT AREA OF SPIRAL BEVEL GEARS UNDER LOAD
(from data by General Motors Research Laboratories)

[4] Spiral Bevel Gear Etching Tests, report from General Motors Research Laboratories, December, 1932 (not published).

2. Experimental Determination of Contact Area

As indicated above, the theoretical calculation of unit loading on gear teeth is dependent upon knowledge of the relative curvatures of the two contacting gear teeth. Because of the complexity of this determination an alternate course is being followed to establish the contact area by experimental means. Upon the establishing of the contact area under varying load conditions, it should then be possible to determine the unit loading on the gear surfaces.

The approach being employed for this portion of the project is the use of the relationship between electrical resistance and contact area of two bodies in contact. This relationship has been shown by Holm[5] as follows:

$$R = \frac{\rho_1 + \rho_2}{4a}$$

Where: R = contact resistance

ρ_1, ρ_2 = resistivity of the materials

a = contact area

Experimental runs are planned to verify this relationship and to determine empirically the values of resistivity of the materials prior to the measuring of contact resistance on hypoid gears. The purpose of these checks is to evaluate the use of this method with a simple system of known contact areas under known loads to determine whether this technique can be used successfully with automotive hypoid gears to measure the area in contact. In order to obtain a wide range of contact area under a relatively limited range of loading conditions, a ball on a flat plate has been

---

5 Holm, R., Electric Contacts, Almqvist and Wiksells, Akademiska Handböcker, 1946.

selected as the system to be used in calibrating this method. The method of calculating the maximum pressure and area of contact for a ball on a flat plate is well established. Using the derivation developed by Prescott and assuming a steel ball 1/2 in. in diameter resting on a flat steel plate, the radius of the contact area and the maximum pressures may be calculated by the following equations:

$$\text{Radius of Contact, } a = \frac{(W)^{1/3}}{652}$$

$$\text{Maximum Pressure, } P\ max = 203{,}000\ (W)^{1/3}$$

Where: W = load on ball, lb.

The derivations of these equations are given in Appendix E. Applying these two formulas, the maximum pressures in contact radii have been plotted in Fig. 42. It will be noted that for a load of 100 lbs. on the ball, a maximum pressure of nearly one million lbs/sq.in. is applied to the center of the contact area. Reducing the load on the ball to 1 lb. reduces the maximum unit pressure to slightly more than 200,000 psi. It, therefore, appears possible to use small loadings on the 1/2 in. diameter ball to obtain pressures in the range which could be expected on gear teeth.

It is apparent that if the electrical contact resistance technique is to be used to determine the area of contact of two hypoid gears under varying loads that the surfaces on the gears must be well polished. It is believed that adequate surfaces can be obtained by a carefully controlled extended run-in of the gears.

D. Future Work Required

In order to accomplish the purposes of this phase of the project, it will be necessary to pursue the following work:

1. Continue the theoretical calculation of unit loading by the developing of a theoretical expression for the curvature of the gear and pinion teeth in the local area of contact.

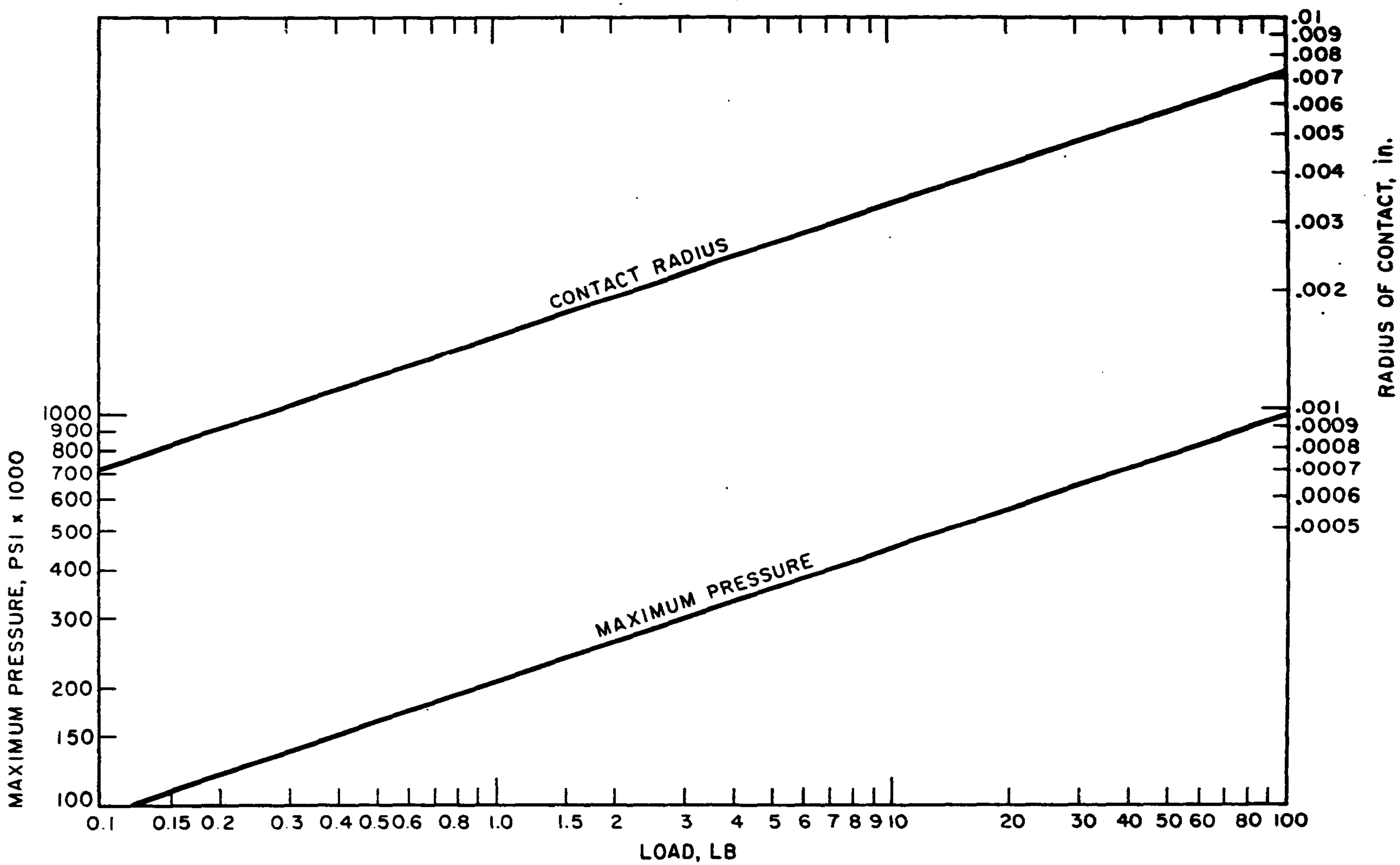


FIG. 42 - MAXIMUM PRESSURE AND RADIUS OF CONTACT OF A HALF INCH DIAMETER STEEL BALL ON A FLAT PLATE.

2. Continue the experimental study of contact area by determining the feasibility of using contact resistance as a means of determining contact area with the experiment of a precision ball on a flat plate. Pending the successful development of this technique, electrical contact resistance and thereby, contact area, of two hypoid gears would be determined under a variety of static loadings.

VI. PHASE V - STUDY OF DYNAMIC SURFACE TEMPERATURE OF HYPOID GEARS

A. Introduction

It has been well established that the lubrication of hypoid gears by the deposition of a chemical film on the surface of the gears is highly dependent upon the operating temperature of the surface of those gears. Most EP materials are relatively inactive at room temperature and somewhat above that value but become increasingly more active at the higher temperatures. Whereas the bulk oil temperature is a good indicator of the relative lubricity of the lubricating oil and the average of the operating temperatures of the unit, it does not indicate directly the temperatures at which the gear surfaces are operating. It is, therefore, considered highly important to the general knowledge of gear lubrication that determinations of operating temperatures of hypoid gears be made.

Whereas it is apparent that the operating temperature of gear surfaces is highly dependent upon both the sliding velocity and the loading of the gear surfaces, it is likewise apparent that other factors such as lubricant or EP film lubricity has a sizable effect upon the surface temperature developed. Thus, it is highly improbable that a single operating temperature is associated with single values of sliding velocities and unit loadings. It is for that reason that it is considered important to determine all three factors for various conditions of gear operation.

A number of investigators have attempted the measurement of surface operating temperatures using several techniques. These techniques have been reviewed as possible means of measuring the surface temperature of hypoid gears. They are listed as follows:

1. Thermocouple Effect of Dissimilar Metal Gears - Several investigators have determined operating contact temperatures for spur gears by using gears cut from dissimilar metals and measuring the thermoelectric potential produced. As compared to most other possible methods, this technique has a great advantage of simplicity. However, it is doubted whether the temperatures developed when using gears cut of dissimilar metals can be accurately related to the surface temperatures developed when gears of identical metals are run together. Further, it is questionable whether the reactions of EP materials on the surfaces of gears manufactured from other than ferrous alloys would produce these same solid films and therefore have the same lubricity or heat transfer characteristics as the film produced on ferrous materials.

2. Plated Thermocouples - Bendersky[6] has developed a high response rate thermocouple of a button of nickel which is plated unto a steel probe. The high response rate is obtained by limiting the thickness of the nickel coating to one micron. A number of investigators have successfully used this technique for the measurement of such high speed phenomenon as the temperature of the inner surface of a gun barrel during firing. The preparation of a plated thermocouple on a gear or pinion tooth presents several very difficult problems.

3. Metallurgical Effects of Temperature - The possibility has been studied of using the metallurgical change in the surface of the gears as a means of determining the temperature generated during contact. The conclusion of this study, carried on in conjunction with several metallurgy specialists, is that this method does not appear attractive since the duration of the elevated temperatures on the surface of the gears is extremely short and the physical effects upon the metallurgy would be very difficult to determine.

4. Infra-red Radiation - The possibility of determining the operating temperatures of gears by the use of infra-red techniques has received some attention. The approach considered was to place an infra-red sensor at a point at which it was in view of the

---

[6] Bendersky, D., "Special Thermocouple for Measuring Transient Temperatures," Mechanical Engineering, V. 75, No. 2, February, 1953.

contacting surfaces of the gears just after they had come out of mesh. Due to the apparent rapid cooling rate of the contact points as they move out of contact and the presence of gear lubricant in both liquid and vapor phases, it was considered highly unlikely that usable information could be obtained in this manner.

5. Color Sensitive Pigments - A series of pigments is available for the determination of operating temperatures of parts and has been considered as a possible method of obtaining gear surface temperatures. However, due to the very low response rate of these pigments, it is apparent that there is no hope of using the available pigments for obtaining high speed temperature transients.

From the methods which have been studied, it appears that the most promising is the use of a plated thermocouple on the surface of the gear teeth, probably located just out of the contact area on the tooth surface. The obvious problems of limited space, plating techniques and the transmitting of the thermocouple signal to a recording device will have to be solved in order to apply this technique successfully.

B. Future Work

The future work on this phase of the project should be applied to the development of a usable technique for experimentally determining temperatures of operating hypoid gears. However, in addition, it is felt that theoretical study should be made in an attempt to develop a mathematical expression for the calculation of surface temperatures of operating gears.

VII. PHASE VI - COOPERATION WITH THE COORDINATING RESEARCH COUNCIL

A. Introduction

In accordance with the policy of the Army, Office of Chief of Ordnance, all test development work undertaken in this project has been in cooperation with industry groups from both the automotive and petroleum industries through the Coordinating Research Council. This cooperation is considered extremely important since any test techniques which are developed under this program are most effective when they are freely accepted by the automotive and petroleum industries as well as government agencies.

During the course of the project on numerous occasions the OCO technical supervisor of the project has requested we cooperate with the Coordinating Research Council in test developments, laboratory investigations and field surveys. A number of these jobs have been associated with phases of the project which have been discussed above and are, therefore, not reported in this phase. The major cooperative programs are discussed in the following section.

B. Discussion of Work Undertaken

1. CRC L-19 Axle Buildup Procedure

The CRC L-19 High Speed Gear Lubricant Test Procedure employs a Chevrolet torque-tube type rear axle assembly. The rebuilding of the test rear axle assemblies used for the L-19 test was accomplished by the Chevrolet Division of General Motors. In 1955 the Chevrolet Division made a major design change in their rear axles. Since the Chevrolet production plants were no longer assembling third members used in the L-19 test, it was necessary to establish a standard technique for use by laboratories conducting these procedures for the disassembly and rebuilding of test axles.

Under this project Armour Research Foundation personnel assumed the responsibility of the chairmanship of a cooperative group charged with the responsibility of developing a standard procedure for rebuilding the L-19 axles. This program was completed in 1956 and the technique developed was used until the replacement of the L-19 procedure with the current L-42 high speed procedure. A copy of the rebuilding procedure is enclosed as Appendix F.

2. <u>Development of CRC L-37 Test</u>

The results of the Army field tests on gear lubricants in heavily loaded trucks in summer operation at the Yuma Test Station in Arizona revealed certain deficiencies in the high torque performance of military gear lubricants. As a result the CRC undertook a program of developing a high torque test which would be usable in evaluating lubricants which would meet the performance requirements of the Yuma tests. Under this phase of the project, cooperative work was undertaken with CRC cooperating laboratories for the development of experimental test techniques. Upon the establishing of a test procedure which produced the proper results, this work was continued with the cooperating laboratories to determine the degree of repeatability and reproducibility of the proposed procedure. At the conclusion of the development of the L-37 test, representatives of the Engines, Fuels and Lubricants Section served on the sub-panel charged with the responsibility for writing the report.

3. <u>Development of CRC L-42 Test</u>

Upon the completion of the development of the L-37 high torque test, several representatives of the automotive industry indicated a dissatisfaction with the current level of anti-scoring performance in multipurpose gear lubricants. As a result of the urging of industry representatives, the CRC undertook a development program for an anti-scoring test technique which would replace the L-19 with a procedure of increased severity. Under this contract, representatives of the Engines, Fuels and Lubricants Section devoted considerable effort to the establishing of the severity level through contacts with the automotive and petroleum industry. In addition, because of the work conducted under this project on the measurement of dynamic torques in automotive gears, the project engineer of this project was appointed leader of the CRC sub-panel on instrumentation for the L-42 technique. As had been the case with the development of the CRC L-37 test, work was undertaken under this project to evaluate experimental techniques and to establish the repeatability and reproducibility of the eventual test procedure developed. At the conclusion of the development of the test technique, the project engineer for this project served as leader of the report writing sub-panel for the CRC.

4. Field Surveys and Investigations

On several occasions during the course of this project, project personnel have been requested to cooperate with the Coordinating Research Council in field surveys to develop data on the required performance level of gear lubricants in automotive equipment, to share in inspections of field test results and to investigate possible service failure reports from military establishments. It is believed that on the whole these surveys and field inspections have been valuable to the Army in providing an independent analysis of the results of the tests or surveys. These inspections and surveys have likewise been valuable to the project personnel in providing them with a familiarity with the field problems and a background for the development of laboratory test techniques.

VIII. ACKNOWLEDGEMENTS

During the course of this project a large number of engineers, scientists and technicians have made valuable contributions to the work. Because of their number, it would be impractical to list them here but reference is made to the progress and summary reports for proper individual credits.

Respectfully submitted,
AUTORESEARCH LABORATORIES, INCORPORATED

D. L. Powell, Project Engineer

APPROVED:

H. R. Barton, President

APPENDIX A

PROPOSED TEST TECHNIQUE FOR DETERMINATION OF MOISTURE CORROSION CHARACTERISTICS OF AUTOMOTIVE GEAR LUBRICANTS

APPENDIX A

RESEARCH TECHNIQUE FOR DETERMINING MOISTURE CORROSION IN REAR AXLE GEAR LUBRICANTS

A. SCOPE

This method describes a procedure for determining the corrosion preventative properties of gear lubricants. The technique is of a severity to duplicate conditions of normal service wherein moisture condenses on metal parts within the gear case under cyclic ambient temperatures. The procedure is applicable to both fresh oil samples and used oil drawn from previously operated gear cases and although developed for extreme pressure lubricants it would also be adaptable to less chemically active mineral base of synthetic lubricants.

B. DEFINITIONS

1. Seven-Day Test - This test consists of the standard 4-hour motoring period followed by a 162-hour storage period. During this storage period the test unit resides in the controlled temperature storage box for 18 hours the first day and for 24 hours the succeeding six days.

2. One-Day Test - This test consists of the standard 4-hour motoring period followed by an 18-hour storage period in the controlled temperature storage box.

C. APPARATUS

The apparatus shall consist of the following:

1. Axle Test Unit - A spicer differential assembly SKA 58391-1X[1].

[1] This unit may be obtained from the Dana Corporation, Toledo, Ohio

2. Test Stand and Driving Apparatus - For description of apparatus required, see Attachment 1, Section A-I.

3. Storage Enclosure - For description of the storage enclosure, see Attachment 1, Section A-II.

4. Temperature Recording and Controlling Equipment - For description of equipment and circuitry required, see Attachment 1, Section A-III.

5. Axle Seals - For description of the axle shaft hole seals and pressure relief valve, see Attachment 1, Section A-IV.

6. Cover Plate Gasket - Victor Gasket No. 27690.

D. PREPARATION OF TEST UNIT

1. Measure pinion torque to break and turn of unit as received. Break and turn torque values must be 15 ± 3 and 10 ± 3 in-lb, respectively.

   NOTE: If torque adjustment is necessary, proceed as follows:

   (a) Remove carrier assembly from housing.

   (b) Check pinion torque. Break torque should be 7 to 10 and turn torque not more than 5 in-lb, respectively.

   (c) To adjust pinion torque add shims to decrease torque or remove shims to increase torque.

   (d) To adjust carrier preload, add shims to increase preload or remove shims to decrease preload.

2. Disassemble test axle.[2]

3. Clean all axle parts with Stoddard Solvent. Carefully inspect all parts for evidence of corrosion. Record location and extent of corrosion present.

   NOTE: Axle unit should be discarded if corrosion is noted on ring or pinion gears or any bearings. New cover plate must be rust-free prior to sandblasting.

---

[2] When conducting a one-day test, a used differential may be used. In this event, spray out the assembly carefully with Stoddard Solvent without disassembling axle and proceed with Step 4.

4. Reassemble axle except for cover plate and install assembly on test stand.

5. Soak a new Victor gasket No. 27690 in a pan of test oil for approximately 5 min.

6. Using approximately one quart of new sand, sandblast a new cover plate until all of the original surface has been removed.[3] This includes the gasket seat surface.

   NOTE: After blasting, avoid handling sandblasted surface and protect from all foreign materials.

7. Remove adherent sand by pouring one quart of Stoddard Solvent over the cover plate. Allow to drain and air-dry. (Spraying of solvent or blowing of air onto cover plate is not recommended since entrained moisture may be present.)

8. Pour small amount of test oil over entire sandblasted area of cover plate and immediately install on differential.

9. Insert axle tube plugs in differential housing (leave one plug loose to provide vent while adding oil and water), and install 3/4 in street elbow in cover plate fill hole.

E. TEST PROCEDURE

1. Fill differential with 2.5 pints of test oil.

2. Start motoring at 2500 rpm and immediately add one ounce (28.3 cc) of fresh distilled water (stored in closed container for not more than two weeks) through the street elbow installed in the cover plate fill hole.

---

[3] The following sand or equivalent is recommended:

Wedron Sand #4098 of 26-Grain Fineness No. (American Foundryman's Soc. Designation). The material is 99.8 per cent SiO_2 with a Moh hardness of 7. This sand is supplied by Wedron Silica Company, 135 South La Salle Street, Chicago, Illinois.

3. Seal street elbow by installing a pipe plug with a pressure relief valve attached (1 lb per sq in opening pressure), and immediately tighten seals in axle tube holes.

4. When oil temperature has reached 180°F (with the aid of two 250-watt heat lamps), install a pipe plug in the relief valve opening to present possible leakage.

5. Motor axle for four hours at 2500 rpm with oil temperature stabilized at 180 ± 2°F (automatic temperature control must be provided to regulate oil temperature by means of heat lamps and a cooling fan).

6. At the completion of motoring, disconnect motor from test assembly and immediately cover differential.

7. Turn on fan, but do not connect strip heaters, and allow to operate until the temperature reaches 140°F (approximately 20-30 minutes).

8. Connect heaters and store differential for time specified at 125 ± 2°F.

   NOTE: At present, tests of one- and seven-days duration are in use. The test sequence is described under Paragraph B - Definitions.

9. At the end of test, drain oil and remove cover plate.

10. Remove unit from test stand. Disassemble unit for inspection if conducting a seven-day test.

F. REPORTING RESULTS

The rating of the lubricant shall be based upon visual inspection for rusting of the differential assembly parts and the axle cover plate inside the gasket facing and above the oil level. The results should be reported as the per cent of the area corroded on the cover plate above the oil level. The intensity and color of the corrosion products should be observed and recorded. In a seven-day test any corrosion on other parts is also to be noted.

A suggested report form is attached as Attachment 2.

ATTACHMENT 1

APPARATUS

A. TEST STAND AND DRIVING APPARATUS

1. The differential assembly is to be supported during the test so that the pinion shaft is horizontal. The differential assembly shall be supported on suitable stands such that it is positioned relative to the table top and the storage box as shown in Drawing No. 1.

2. During the motoring phase of the test, the two 250-watt heating lamps and the cooling fan shall be positioned relative to the test axle as shown in Drawing No. 2.

3. A suitable motor support and pulley and belt system shall be provided for driving the test unit by an electric motor of approximately 1-1/2 HP rating at 2500 rpm. Drawing No. 3 shows a suggested arrangement for this apparatus.

B. STORAGE ENCLOSURE

1. During the storage phase of the test the differential assembly is enclosed in an air tight temperature controlled box which provides uniform circulation of air.

2. The storage enclosure is an aluminum box enclosing an aluminum baffle with a centrifuged blower located at the top center of the box. Four electric resistance heaters are mounted along the lower edges of the baffles such that after the air circulates between the outer box and the inner baffle it must pass below the heaters. The details of construction of the storage box and its wiring are shown In Figs. 4 through 11.

3. The following assembly procedure is provided to assist in fabricating and installing this equipment.

(a) Fabricate the outer box, inner baffle, heater mounting brackets and the fan in accordance with Drawings 4 thr ugh 7.

(b) Wire toggle switch and receptacles. One male and two female plugs are attached to the receptacles with the spring clip provided.

(c) Install toggle switch and receptacles, mounting the receptacle with male plug in the hole on the side of the box and the two female plugs in the holes on the top of the box.

(d) Wire receptacles and switch as shown in Fig. 8. Because the only access to the receptacles after the box has been assembled will be from the outside, sufficient slack should be allowed in wiring to enable the receptacles to be pulled out.

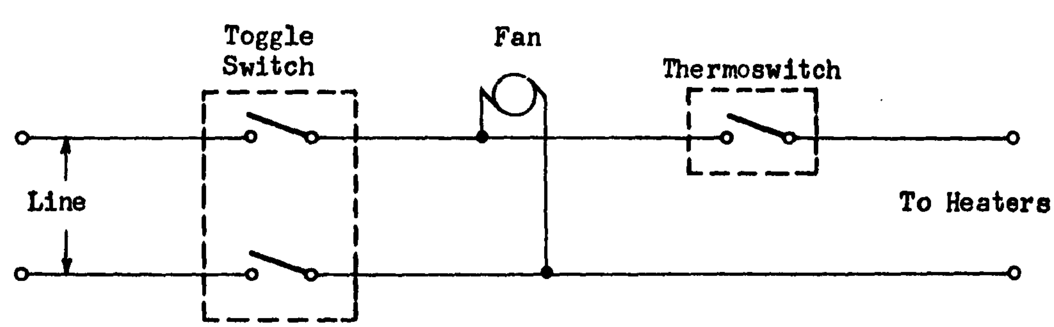


Fig. 8 Wiring Diagram

(e) Attach two pieces of wire approximately four feet long to the receptacles for leads to the heaters.

(f) Attach one male plug to motor lead wires.

(g) Mount motor on top of box providing sufficient spacers on the mounting screws to insure that the motor housing is not forced against the box causing the bearing to jam. Fasten the motor to the box with two nuts.

(h) Install fan impeller on motor shaft with set screw.

(i) Push inner baffle into outer box until bottom edge of baffle is approximately flush with the bottom edge of the box.

NOTE: These two parts should fit snugly and require light force to assemble.

(j) Attach the heater to the mounting brackets and secure the brackets to the inner baffle as shown in Fig. 9.

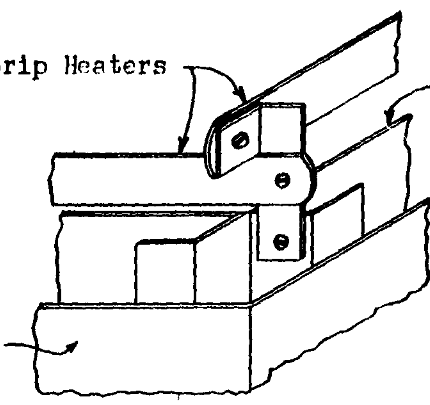


Fig. 9 Strip Heater Mounting Detail

(k) Wire the heaters together and to the receptacles as shown in Fig. 10.

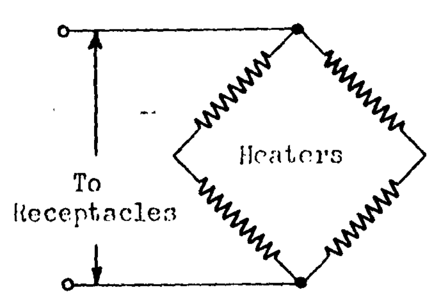


Fig. 10 Heater Wiring Diagram

(l) Force the inner baffle into the outer box until the top of the baffle is flush with the outer edge of the impeller blades.

(m) Plug in power cord and check operation of fan to insure that no interference exists between the shaft or the impeller and the box or baffle.

(n) Secure the inner baffle in place by drilling holes through the box and the web of the baffle and bolt the two parts together. (Two bolts should be adequate.)

(o) With the inner baffle in position drill a 5/8-inch hole in the inner baffle to match the thermoswitch mounting hole in the outer box.

(p) Mount the thermoswitch in place by drilling and tapping two screw holes in the outer box.

(q) Wire the thermoswitch with approximately 1-1/2 foot leads and a male plug.

(r) Clamp the bottom seal to the outer box with aluminum strips as shown in Fig. 11.

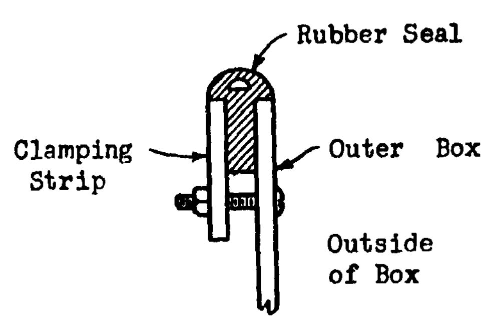


Fig. 11 Seal Mounting Detail

4. The following is a list of parts required for the construction of the storage enclosure.

   (a) Outer box - Fig. 4.

   (b) Inner baffle - Fig. 5.

   (c) 4 - Heater mounting brackets - Fig. 6.

   (d) Fan impeller - Fig. 7.

   (e) 1 - Heavy duty DP-ST toggle switch such as Cutler-Hammer No. 7560-K5.

   (f) 3 - Receptacles - Amphenol #61-61.

   (g) 1 - Male plug - Amphenol #61M.

   (h) 2 - Female plugs - Amphenol #61F.

   (i) 2 - Male connectors - Amphenol #61M11.

   (j) 1 - Female connector - Amphenol #61F11.

   (k) Fan motor - Dayton Electric Company, 1/30 HP Model 4K102, 1550 rpm.

   (l) 4 - Chromalox strip heaters No. S2450, 110 V, 500 W.

   (m) Fenwall thermoswitch, Cat. No. 17752.

   (n) 12 ft Jarrow Products refrigerator gasket, Cat. No. 2420.

   (o) 4 - Strips 1/8 inch aluminum plate, 24 in. x 1 in.

   (p) Assorted nuts and bolts for attaching fixtures.

C. TEMPERATURE RECORDING AND CONTROLLING EQUIPMENT

1. The temperature of the lubricant in the differential assembly to be measured and continuously recorded by means of a suitable instrument. A Foxboro resistance type Model E-742 temperature probe used in conjunction with a Foxboro Dynalog Model 9135W-M2-"754SI-11" is satisfactory.

2. The temperature probe is to be mounted in the differential housing through the drain plug so that at least one inch of the probe is immersed in the lubricant and the sensitive portion of the probe is approximately 1/2 inch from the ring gear face.

3. During the motoring phase of the test the temperature of the lubricant is maintained at 180 $\pm$ 2°F by means of two 250-watt heat lamps and a fan. The heat lamps and fan operate alternately switched automatically by the recorder-controller.

4. The temperature of the lubricant in the differential assembly is maintained at 125 $\pm$ 2°F during the storage period of the test. This temperature is maintained by controlling the circulating air temperature in the box by means of the control system described in Paragraph B.

D. <u>AXLE SEALS AND RELIEF VALVE</u>

1. The axle shaft holes of the differential assembly shall be sealed during the test after the addition of the water. Drawing No. 12 shows the details of the suggested plugs.

2. The differential assembly is vented during the motoring phase of the test through a pressure relief valve set to open at 1 psi. A James Bond Clark Model 259B-2PP relief valve set for 1 psi opening pressure is suggested.

APPENDIX B

CRC L-19 TEST

TABULATED DATA OF LABORATORY & FIELD TORQUE RECORDINGS

TABLE B-I

MAXIMUM RING GEAR SHOCK TORQUES FROM L-19 TEST

Shock	Ring Gear Torque lb/ft Road Test			Laboratory Test
	Carburetor No. 1	Carburetor No. 2	Carburetor No. 3	Carburetor No. 3
10 mph * Drive Side	960 1150	1220 1455	1237 1033	562 613 682 732
60 mph* Drive Side	613 717 1090	784 960	733 1182 1237	906 922 1015
40 mph** Coast Side	550 580	550 630	580 770	650 682 805 837
80 mph ** Coast Side	600 665 750	820 980	500 630 870	770 783

* Major shock loads at 10 mph and 60 mph are on the drive side of the gears.
** Major shock loads at 40 mph and 80 mph are on the coast side of the gears.

TABLE B-II

MAXIMUM RING GEAR SHOCK TORQUES

FROM G.M. SHOCK TEST

Shock	Ring Gear Torque, lb/ft Road Test (Carburetor No. 1)
65 mph (High Gear)	2060 2185
45 mph (Second Gear)	4193 4280 4325

APPENDIX C

TABULATED DATA OF GEAR LOADINGS IN

ARMY FIELD TESTS

TABLE C-1

AVERAGE AXLE TORQUES ON M-37 VEHICLE OVER YUMA HIGHWAY PORTION OF TEST COURSE

	WITH TRAILER			WITHOUT TRAILER		
	Axle[1] Torque lb-ft	Speed mph	Gear Range	Axle[1] Torque lb-ft	Speed mph	Gear Range
Maximum Steady Load on Hill at North End of Course						
Uphill	3385 d	24.0	H-3	2280 d	37.5	H-4
Downhill	90 c	46.5	H-4	2885 c	48.2	H-4
Typical Bumps in Pavement	443 c 1884 d	44.6	H-4	-	-	-
Typical Dip in Pavement	1580 d	43.0	H-4	-	-	-
Steady Speed Level Road Load	93 d	15.5	H-4	60 d	15.4	H-4
	258 d	30.1	H-4	78 d	30.4	H-4
	332 d	40.2	H-4	150 d	39.9	H-4

TABLE C-2

AVERAGE AXLE TORQUES ON M-37 VEHICLE OVER DEATH VALLEY PORTION OF TEST COURSE

	WITH TRAILER			WITHOUT TRAILER		
	Axle[1] Torque lb-ft	Speed mph	Gear Range	Axle[1] Torque lb-ft	Speed mph	Gear Range
Average Axle Torques:						
Daylight Pass						
Uphill	1950 d	16.7	L-3	1250 d	27.2	L-4
Downhill	200 c	27.5	L-4	140 c	30.5	L-4
Towne Pass						
Uphill	1500 d	16.3	L-3	1300 d	25.5	L-4
Downhill	140 c	40.9	H-4	940 c	23.0	L-4
Maximum Steady Axle Torques:						
Maximum Slope on Daylight Pass						
Uphill	3750 d[2]	14.0	L-3	1020 d	21.3	L-4
Downhill	100 d[2]	18.5	L-3	2050 c	30.0	L-4
Maximum Slope on Towne Pass (East Side)						
Uphill	2170 d	14.8	L-3	3675 d[3]	16.5	L-3
Downhill	375 c	20.5	L-3	2300 c[3]	27.2	L-4
Maximum Slope on Towne Pass (West Side)						
Uphill	4425 d	8.5	L-3	3310 d	15.3	L-3
Downhill	1300 c	20.0	L-2	300 c	30.0	L-4

(1) d Indicates loading on drive side of gears, c Indicates loading on coast side of gears.
(2) Brakes applied steadily to both trailer and truck.
(3) Accelerating rapidly.

TABLE C-3

AVERAGE AXLE TORQUES ON M-211 VEHICLE OVER YUMA HIGHWAY PORTION OF TEST COURSE

	WITH TRAILER				WITHOUT TRAILER			
	Torque Intermed. Axle(1) lb-ft	Torque Rear Axle(1) lb-ft	Gear Range	Speed mph	Torque Intermed. Axle(1) lb-ft	Torque Rear Axle(1) lb-ft	Gear Range	Speed mph
Maximum Steady Load On Hill at North End of Course								
5500 lb Trailer								
Uphill	1400 d	1060 d	H-2	17.7	1130 d	950 d	H-2	18.7
Downhill	100 c	325 c	H-4	49.4	300 c	325 c	H-4	51.2
7825 lb Trailer								
Uphill	1780 d	1200 d	H-2	18.2	-	-	-	-
Downhill	125 c	240 c	H-4	49.4	-	-	-	-
Typical Bumps in Pavement								
5500 lb Trailer	798 d	426 d	H-4	50.7	748 d	428 d	H-4	54.2
7825 lb Trailer	829 d	281 d	H-4	43.9	-	-	-	-
Typical Dip in Pavement								
5500 lb Trailer	565 d	180 d	H-4	49.9	479 d	132 d	H-4	54.8
7825 lb Trailer	559 d	140 d	H-4	42.7	-	-	-	-

TABLE C-4

AVERAGE AXLE TORQUES ON M-211 VEHICLE OVER DEATH VALLEY PORTION OF TEST COURSE

	WITH TRAILER (2)				WITHOUT TRAILER			
	Torque Intermed. Axle(1) lb-ft	Torque Rear Axle(1) lb-ft	Gear Range	Speed mph	Torque Intermed. Axle(1) lb-ft	Torque Rear Axle(1) lb-ft	Gear Range	Speed mph
Average Axle Torques:								
Daylight Pass								
Uphill	360 d	410 d	L-4	1[illegible].0	375 d	360 d	H-3	19.2
Downhill	175 c	210 c	H-3	18.7	135 c	160 c	H-3	27.2
Towne Pass								
Uphill	1850 d	1500 d	L-4	13.8	1075 d	925 d	L-4	16.4
Downhill	920 c	800 c	L-4	14.3	750 c	750 c	L-4	11.3
Maximum Steady Axle Torques:								
Maximum Slope Daylight Pass								
Uphill	3240 d	3010 d	L-2	8.3	2000 d	2000 d	L-4	11.3
Downhill	1570 c	850 c	L-4	13.3	750 c	640 c	H-3	21.0
Maximum Slope Towne Pass (East Side)								
Uphill	1990 d	1970 d	L-3	9.9	1320 d	1070 d	L-4	13.0
Downhill	1075 c	510 c	L-4	13.8	200 c	440 c	H-2	16.0
Maximum Slope Towne Pass (West Side)								
Uphill	1075 d	1225 d	L-3	8.4	1140 d	1160 d	L-3	10.1
Downhill	380 c	630 c	L-4	13.0	250 c	300 c	H-3	21.6

(1) d Indicates loading on drive side of gears, c Indicates loading on coast side of gears.
(2) Trailer Gross Weight - 5500 lbs.

TABLE C-5

AVERAGE GEAR SHOCK TORQUES IN M-211 TRUCK DUE TO GEAR SHIFTING

Gear Shifts	Slope of Road	Ring Gear Torque,(1) lb-ft Intermediate Axle	Rear Axle	Speed mph
High Gear Range:				
From High 1 to High 2	Level	2875 d	2275 d	14.3
From High 2 to High 3	Level	3060 d	2450 d	21.6
	Uphill	2825 d	2278 d	22.0
From High 4 to High 3	Level	2910 c	2130 c	10.2
From High 3 to High 2	Uphill	2253 c	2223 c	16.0
		2983 d	2363 d	
From High 2 to High 1	Level	1630 c	1150 c	7.7
Low Gear Range:				
From Low 3 to Low 4	Uphill	3763 d	3483 d	9.6
From Low 4 to Low 3	Uphill	4828 d	4534 d	8.5
Shifting Between High and Low Range:				
From Low 4 to High 2	Uphill	2300 d	1625 d	13.0
From High 2 to Low 4	Downhill	5650 c	5045 c	7.2

TABLE C-6

AVERAGE GEAR SHOCK TORQUES IN M-37 TRUCK DUE TO GEAR SHIFTING

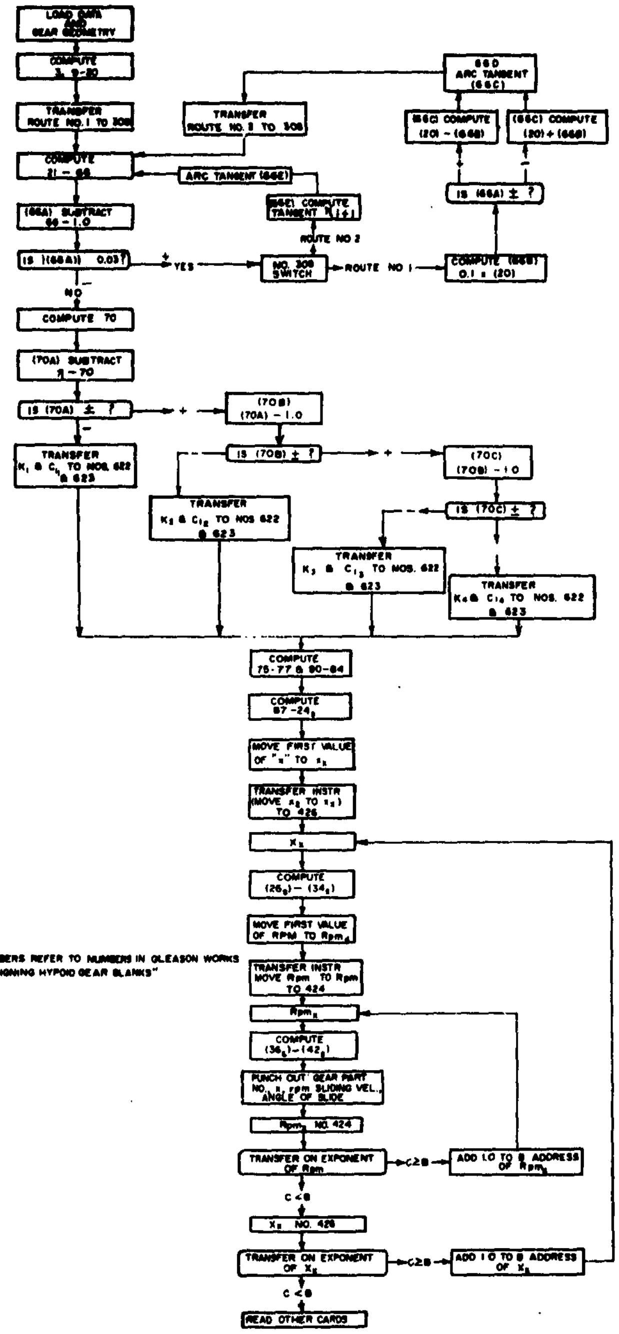
Gear Shifts	Slope of Road	Ring Gear Torque,(1) lb-ft	Speed mph
Low Gear Range:			
From Low 3 to Low 4	Uphill	635 c	22.3
		3410 d	
	Downhill	1280 c	19.3
From Low 4 to Low 3	Uphill	2095 c	19.5
		5510 d	
	Downhill	3475 d	19.5
		1010 c	

(1) d Indicates loading on drive side of gears, c Indicates loading on coast side of gears.

APPENDIX D

BLOCK DIAGRAM OF DIGITAL COMPUTER PROGRAM

FOR HYPOID GEAR SLIDING COMPUTATION



NOTE: CALCULATION NUMBERS REFER TO NUMBERS IN GLEASON WORKS "METHOD FOR DESIGNING HYPOID GEAR BLANKS"

HYPOID GEAR SLIDING VELOCITY CALCULATION IBM650 COMPUTER PROGRAM

APPENDIX E

DERIVATION OF EXPRESSION FOR RADIUS OF CONTACT & MAXIMUM PRESSURE BETWEEN A SPHERE & A FLAT PLATE

APPENDIX E

DERIVATION OF THE EXPRESSION FOR RADIUS OF CONTACT AND MAXIMUM PRESSURE BETWEEN A SPHERE AND A FLAT PLATE

Suppose two bodies A and B are in contact at a point O with no pressure between them. Let OZ_1 be the normal to the surface of the body A, the direction of OZ_1 being toward the inside of the body. If x, y, z, are the coordinates, referred to rectangular axes OX, OY, OZ of a point on the surface of A, the equation to the surface in the immediate neighborhood of O is shown to be

$$z_1 = a_1x^2 + b_1y^2 + 2h_1xy \tag{1}$$

a_1, b_1, h_1 being constants.

Again if OZ_2 is the normal to the surface of the body B, the direction OZ_2 being toward the inside of B, the equation to the surface of B in the immediate neighborhood of O is

$$z_2 = a_2x^2 + b_2y^2 + 2h_2xy \tag{2}$$

Let

$$z = z_1 + z_2$$

Substituting Eq. (1) and (2):

$$z = (a_1 + a_2)x^2 + (b_1 + b_2)y^2 + 2(h_1 + h_2)xy \tag{3}$$

This is the relative indicatrix of the two surfaces at O.

If the relative indicatrix is a circle, then

$$h_1 + h_2 = 0,$$

and $$a_1 + a_2 = b_1 + b_2$$

Let $$k = a_1 + a_2 = b_1 + b_2 \tag{4}$$

Then Eq. (3) for the relative indicatrix becomes

$$z = k(x^2 + y^2) = kr^2 \qquad (5)$$

If a pressure acts over a circle of radius a#, having its center at the origin, the pressure at radius r is given by

$$p = C(a^2 - r_1^2)1/2 \qquad (6)$$

The displacement, w, at (x, y, o) due to a distributed load may be shown to be

$$\frac{1 - \sigma}{2\pi n} \iint \frac{p\, dx_1\, dy_1}{R}$$

where σ = poisson's ratio
n = shear modulus

Substituting and solving for w:

$$w = \frac{(1 - \sigma) \quad C}{4n} \quad (a^2 - \frac{r^2}{2}) \qquad (7)$$

The total force is then

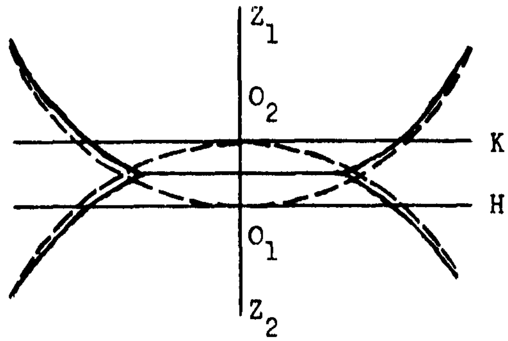
$$w = \int p\, dx_1\, dy_1 = \int_o^a 2\pi r_1\, pdr_1 = \frac{2\pi}{3} Ca^3 \qquad (8)$$

Now assume two bodies are pressed together and that the surface of contact is a circle of radius a. A spherical depression is made in each body and the normal displacements in the two bodies within the area of contact are:

$$w_1 = \frac{(1 - \sigma_1)}{4n_1} \quad (a^2 - \frac{r^2}{2}) \qquad (9)$$

$$w_2 = \frac{(1 - \sigma_2)}{4n_2} \quad (a^2 - \frac{r^2}{2}) \qquad (10)$$

where w_1 is measured from the plane O_1H in the direction of $O_1\ Z_1$ and w_2 is measured from the plane O_2K in the opposite direction.



The constant C is the same for both bodies since the pressure p is the same for both.

The distance of a point on the strained surface of A from the plane O_1H is $(z_1 + w_1)$, and the distance of a point of the strained surface of B from O_2K is $(z_2 + w_2)$. If d denotes the distance between the two planes O_1H and O_2K we must have in the area of contact

$$(z_1 + w_1) + (z_2 + w_2) = d$$

$$w_1 + w_2 = d - (z_1 + z_2)$$

$$= d - kr^2$$

or, using (8) to express C in terms of total force, W,

$$\frac{3W}{8\pi a^3} \quad \frac{1-\sigma_1}{n_1} + \frac{1-\sigma_2}{n_2} \quad (a^2 - \frac{r^2}{2}) = d - kr^2 \qquad (11)$$

Since this is true for a value or r less than a, we get

$$\frac{3W}{8\pi a} \quad \frac{1-\sigma_1}{n_1} + \frac{1-\sigma_2}{n_2} = d$$

and

$$\frac{3W}{16\pi a^3} \quad \frac{1-\sigma_1}{n_1} + \frac{1-\sigma_2}{n_2} = k \qquad (12)$$

From (6) the maximum pressure may be seen to be

$$P_{max} = Ca = \frac{3W}{2\pi a^2} \qquad (13)$$

Particular Example of a Sphere on a Plane

Suppose a steel ball with a diameter of half an inch is pressed against a large steel plate.

Assume that n and σ have the same values for both bodies as follows:

$$n_1 = n_2 = 11,500,000$$

$$\sigma_1 = \sigma_2 = 0.303$$

The value k denotes half the sum of the curvatures of the sections of the two bodies by any plane containing the common normal. In this case k is half the curvature of the sphere.

From Eq. (12)

$$a^3 = \frac{3\,(1 - \sigma_1)}{8\,\pi} \quad \frac{W}{kn_1}$$

$$a^3 = \frac{W}{277,500,000}, \quad a = \frac{(W)^{1/3}}{652}$$

and

$$P_{max} = \frac{3W}{2\,\pi a^2} = 203,000\ (W)^{1/3}.$$

APPENDIX F

A STANDARD TECHNIQUE FOR REBUILDING CHEVROLET TEST AXLES FOR THE CRC L-19 TEST

A STANDARD TECHNIQUE FOR REBUILDING CHEVROLET

TEST AXLES FOR THE CRC L-19 TEST

I. **PURPOSE**

The purpose of these instructions is to provide a standard procedure for the rebuilding of Chevrolet third member assemblies for use in CRC L-19 High Speed Gear Lubricant Tests. Care should be taken to follow the instructions explicitly, if erroneous test results are to be avoided.

II. **STANDARDIZATION RUNS**

A. Third Member Assembly

Each third member assembly when new, and after 20 runs, is to be standardized by running in a complete CRC L-19 Test using a borderline reference oil.* The test must produce some scoring on the ring and pinion gears. If these results are not obtained, the assembly should not be used for testing.

B. Test Gear Lots

Test gears should be purchased in reasonable quantity from one lot if possible, conducting one reference test using the reference oil and producing the results specified in paragraph IIA.

III. **PARTS INSPECTION**

A. New Axle, or After 20 Runs

1. Propeller Shaft - Replace if found defective

   a. Check for straightness by placing on a flat surface and rolling slowly, observing any lack of straightness. (Procedure for straightening to be developed)

   b. Check for balance. (Procedure to be developed)

   c. Inspect splines for excessive wear or looseness.

   d. Inspect coupling pin for looseness.

   e. Inspect shaft at propeller shaft bushing location for scoring or excessive wear.

* Borderline reference oils for use in axle standardization have not been established. It is suggested that laboratories select an oil for this purpose until suitable oils are developed by the CRC Panel.

2. Torque Tube Assembly

   a. Inspect pinion bearing outer races for looseness in housings.

   b. Inspect side carrier bearing for looseness in housings.

   c. Inspect front bushing for excessive wear, replace if necessary.

3. Differential Carrier Assembly

   a. Inspect ring gear mounting flange for nicks or burrs.

   b. Inspect side carrier bearing cones and rollers for excessive wear or surface damage.

B. Each Test

1. Gears

   a. Check tooth hardness with 61 Rockwell C Test file. If tooth surface is cut by file do not use gear.

   b. Inspect for surface irregularities such as nicked or chipped teeth or evidence of grinding at top of teeth.

2. After cleaning and oiling bearings, check for roughness by slowly turning the outer race by hand.

3. Check clearance between propeller shaft and its bushings. If this clearance exceeds 0.010 in. the bushing and oil seal should be replaced.

IV. <u>ASSEMBLY INSTRUCTIONS</u>

A. The replacement parts required for rebuilding axles are listed in Supplement I.

B. Special tools required are listed in Supplement II.

C. Assembly Instructions

1. Preparation of sub-assemblies

   a. To replace propeller shaft bushings and oil seal:

      1) Remove bushings and oil seal from housing by using the special puller (J4258), or by driving from pinion end.

      2) Start a new seal into housing with free side of leather toward front.

3) Install new rear bushing and drive new bushing and seal firmly against their seat with bushing driver (J-968).

4) Install front bushing using bushing driver (J-968).

b. To assemble pinion and propeller shaft:

1) Install rear pinion bearing on pinion shaft and lock in place with lock ring.

2) Install pinion bearing lock sleeve on shaft with beveled edge toward the pinion.

3) Press front (double ball) bearing on shaft and install bearing lock nut.

4) Install coil spring over end of spline.

5) Install splined end of pinion shaft into coupling on end of propeller shaft aligning rivet holes.

6) Install new rivets, and rivet both ends.

7) Tighten bearing lock nut to 200 - 240 ft-lbs and lock in milled slot in pinion shaft.

c. Differential Carrier Assembly

1) After each failing test, or after each fifth passing test, replace the side carrier bearings as follows:

a) Install differential bearing puller TR 278-R, making sure puller legs are fitted securely in notches in case and retaining yoke tight.

b) Tighten puller screw and remove bearing.

c) Replace bearing by placing on hub with the thick side of inner race toward case.

d) Drive bearing in place with differential side bearing replacer J-994.

2) Ring Gear Replacement

a) Remove ring gear bolts and lockwashers.

b) With soft hammer tap ring gear off the case.

c) Install guide pins made from 3/8" - 24 x 1-1/2" long capscrew with heads cut off and ends slotted to new ring gear.

d) Make sure back face of ring gear and face of case are free of dirt and burrs, and slip gear over pilot diameter of case.

e) Install every other ring gear bolt and lockwasher, then draw them up evenly and snugly so that ring gear face is flush with face of case.

f) Remove guide pins and install remaining bolts.

g) Tighten all bolts to 40-60 lb-ft.

h) Install differential carrier into axle housing and check run-out of rear face of ring gear. This run-out not to exceed 0.0035 in.

2. Complete Axle Assembly

a. Propeller Shaft Installation

1) Place one 0.018 in. shim in the propeller shaft housing counterbore.

2) Install propeller shaft and pinion assembly, driving it down until bearings are seated in the housing using spacer tool J-4050 to provide proper pinion to bearing clearance.

3) Check through bearing lock screw holes to make sure lock sleeve is in position against back of front pinion bearing.

4) Install three tapered lock screws and draw them down evenly and tightly, then tighten lock screw nuts.

b. Differential Carrier Installation

1) Install differential assembly in the carrier and install adjusting nuts.

CAUTION: CAREFULLY SLIDE ADJUSTING NUTS ALONGSIDE THE BEARING SO THAT THREADS ON NUTS FIT INTO THREADS IN CARRIER.

2) Install bearing caps, aligning marks on cap with marks on carrier.

3) Install and tighten cap screws until lock washers just flatten out.

c. Gear Contact Adjustment

1) Loosen right-hand adjusting nut and tighten left-hand adjusting nut, using differential adjusting wrench J-972, while turning ring gear. Continue tightening left-hand nut until all lash is removed, then back off the left-hand nut one notch to a locking position.

2) Tighten right-hand nut to force left bearing firmly into contact with left adjusting nut. Then loosen the right nut and again tighten snugly against the bearing.

NOTE: THIS POSITION MAY BE EASILY DETERMINED AS THE NUT COMES TO A DEFINITE STOP.

3) Tighten right-hand nut a minimum of one additional notch, to a maximum of two notches further, to a locking position. This operation preloads the differential bearings.

4) Mount a dial indicator on the carrier, and with the pinion locked, check the back-lash between the ring gear and pinion gear teeth. Back-lash must be between 0.007 in. and 0.012 in.

5) To check proper tooth contact:

a) Wipe ring gear and pinion dry with clean cloth and paint teeth lightly and evenly with red lead of suitable consistency.

b) Apply a load on pinion to provide drag.

c) Turn ring gear over slowly, then check pattern of tooth contact.

d) Contact at toe of teeth (See Fig. I-B) indicates insufficient back-lash. To correct, loosen left-hand differential adjusting nut and tighten right-hand adjusting nut. Make adjustments one notch at a time, checking with red lead as above.

e) Contact at heel of teeth (See Fig. I-C) indicates too much back-lash. Adjust as in d) above, except loosen right-hand nut and tighten left-hand nut.

f) Contact at face of teeth (See Fig. I-D) indicates insufficient shim thickness behind pinion. To correct, remove differential assembly and pinion and propeller shaft. Measure shim thickness and use new combination of shims to provide approximately 0.006 in. increase. Reassemble unit and repeat check with red lead. Continue adjustment until the tooth contact is normal.

g) Contact at flank of teeth (See Fig. I-E) indicates too many shims behind pinion. Disassemble unit and reduce shim thickness about 0.006 in., reassemble, and repeat check.

h) In making pinion adjustments, check back-lash before testing with red lead. Moving the pinion in or out changes the back-lash.

i) After tooth contact is correct, wipe red lead from gears and carrier with cloth moistened with clean gasoline or kerosene.

3. Installation of Carrier in Axle Housing

a. Clean out axle housing and cover, and place new gasket over carrier mounting bolts.

b. Assemble carrier to axle housing, install lock washers and nuts, and tighten securely.

c. Install axle shafts, longer shaft on right side and install "C" shaped axle shaft locks.

d. Spread shafts to make sure that the shafts, locks and differential side gears are in positive contact.

e. Roll the two differential pinions into position, install axle shaft spacer, pinion gear shaft and pinion gear shaft lock screw.

f. Check clearance between end of axle shaft and spacer. This should be from free fit to 0.014 in.

g. Install axle housing cover using new gasket.

h. Raising transmission end of torque tube, pour approximately 1/4 pint of test gear lubricant into end of torque tube to lubricate torque tube bushings and seal.

i. Assemble propeller shaft to transmission by connecting universal joint and assemble universal ball joint.

V. INSPECTIONS DURING TEST

A. Purpose of these inspections is to maintain a continuing check on assembly techniques and replacement part quality.

B. Remove inspection cover and observe condition of ring gear at following intervals:

1. After run-in

2. After 10 - 40 mph accelerations

3. At end of test

VI. DISASSEMBLY AFTER TEST

A. Removal of carrier from axle housing.

1. Disconnect universal joint from propeller shaft at transmission.

2. Remove rear cover from axle housing. Remove differential pinion shaft and spacer block.

3. Push axle shafts in toward differential until "C" shaped axle shaft locks are free. Remove locks and pull axle shafts out of differential.

4. Remove nuts and lock washers holding carrier to axle housing and remove carrier from axle.

B. Disassembly of Differential Carrier Assembly.

1. Remove adjusting nut locks and four differential carrier cap screws.

2. Remove bearing caps and adjusting nuts, and remove differential assembly.

3. Replace ring gear as described in Paragraph IV C 1c 2).

4. After each failing test, or each five passing tests, replace side carrier bearings as described in Paragraph IV C 1c 1).

C. Removal and Disassembly of Pinion and Propeller Shaft.

1. Remove three tapered bearing retaining screws and tap splined end of propeller shaft allowing pinion and shaft to slide out.

2. Remove shims from inside of propeller shaft housing, noting number and total thickness of shims removed as an aid in rebuilding the axle.

3. Drill end of coupling pin to clear countersink into which it is upset, being careful to center the rivet with a center punch.

4. Drive out coupling pin.

5. Loosen pinion bearing lock nut and separate pinion from propeller shaft.

6. After each gear failure, or each five passing tests, replace pinion bearings as follows:

    a. Remove pinion bearing lock nut and press bearing from pinion shaft using pinion bearing remover J-996.

    b. Remove bearing lock sleeve and rear bearing lock ring.

    c. Remove rear bearing from pinion shaft.

SUPPLEMENT I

The parts most frequently replaced in the L-19 Tests are listed below:

A. PARTS NEEDED AS REQUIRED

ITEM	PART NO.
Torque Tube Oil Seal	3694736
Torque Tube Bushing, front	3694732
rear	3694733
Propeller Shaft Counterbore shims:	
0.012"	3657740
0.015"	3657741
0.018"	3657742
0.021"	3657743

B. PARTS REQUIRED EVERY TEST

Ring Gear) Pinion) (Purchase as set)	3705430
Pinion Coupling Pin	3657332
Gasket, front	370246
rear	370245
Side Carrier Bearings (if gears scored)	127861
Pinion Bearings, front "	954780
rear "	125630

C. PARTS REQUIRED EVERY FIVE TESTS

Side Carrier Bearings	127861
Pinion Bearings, front	954780
rear	125630

NOTE: Parts presently available through local Chevrolet parts dealer.

AD______________________________ Accession No.______________

Conducted and Prepared by AutoResearch Laboratories, Incorporated, Chicago, Illinois for Armour Research Foundation, Chicago, Illinois. Final Summary Report on DEVELOPMENT OF QUALIFICATION TEST METHODS FOR GEAR LUBRICANTS. D. L. Powell

April, 1961, (142) p (Contract DA-11-022-ORD-905) DA Proj. -593-21-060, OCO Project TB5-2, Unclassified Report.

A multi-phase experimental and analytical project has been undertaken to develop test procedures for the evaluation of hypoid gear lubricants and to study the loading, temperature and sliding velocity conditions which exist on automotive hypoid gears. Each of the six phases of this work are discussed in individual sections of this report.

The work included in this project conducted in the period from 1952 to 1960 is as follows:

- I. Development of Moisture Corrosion Test Technique
- II. Study of Dynamic Loading of Automotive Hypoid Gears
- III. Study of Sliding Velocity of Automotive Hypoid Gears
- IV. Study of Unit Surface Loading of Hypoid Gears
- V. Study of Dynamic Surface Temperatures of Hypoid Gears
- VI. Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations.

UNCLASSIFIED

1. Procedure development - Gear lubricants
2. Gear Lubrication

AD______________________________ Accession No.______________

Conducted and Prepared by AutoResearch Laboratories, Incorporated, Chicago, Illinois for Armour Research Foundation, Chicago, Illinois. Final Summary Report on DEVELOPMENT OF QUALIFICATION TEST METHODS FOR GEAR LUBRICANTS. D. L. Powell

April, 1961, (142) p (Contract DA-11-022-ORD-905) DA Proj. -593-21-060, OCO Project TB5-2, Unclassified Report.

A multi-phase experimental and analytical project has been undertaken to develop test procedures for the evaluation of hypoid gear lubricants and to study the loading, temperature and sliding velocity conditions which exist on automotive hypoid gears. Each of the six phases of this work are discussed in individual sections of this report.

The work included in this project conducted in the period from 1952 to 1960 is as follows:

- I. Development of Moisture Corrosion Test Technique
- II. Study of Dynamic Loading of Automotive Hypoid Gears
- III. Study of Sliding Velocity of Automotive Hypoid Gears
- IV. Study of Unit Surface Loading of Hypoid Gears
- V. Study of Dynamic Surface Temperatures of Hypoid Gears
- VI. Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations.

UNCLASSIFIED

1. Procedure development - Gear lubricants
2. Gear Lubrication

AD______________________________ Accession No.______________

Conducted and Prepared by AutoResearch Laboratories, Incorporated, Chicago, Illinois for Armour Research Foundation, Chicago, Illinois. Final Summary Report on DEVELOPMENT OF QUALIFICATION TEST METHODS FOR GEAR LUBRICANTS. D. L. Powell

April, 1961, (142) p (Contract DA-11-022-ORD-905) DA Proj. -593-21-060, OCO Project TB5-2, Unclassified Report.

A multi-phase experimental and analytical project has been undertaken to develop test procedures for the evaluation of hypoid gear lubricants and to study the loading, temperature and sliding velocity conditions which exist on automotive hypoid gears. Each of the six phases of this work are discussed in individual sections of this report.

The work included in this project conducted in the period from 1952 to 1960 is as follows:

- I. Development of Moisture Corrosion Test Technique
- II. Study of Dynamic Loading of Automotive Hypoid Gears
- III. Study of Sliding Velocity of Automotive Hypoid Gears
- IV. Study of Unit Surface Loading of Hypoid Gears
- V. Study of Dynamic Surface Temperatures of Hypoid Gears
- VI. Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations.

UNCLASSIFIED

1. Procedure development - Gear lubricants
2. Gear Lubrication

AD______________________________ Accession No.______________

Conducted and Prepared by AutoResearch Laboratories, Incorporated, Chicago, Illinois for Armour Research Foundation, Chicago, Illinois. Final Summary Report on DEVELOPMENT OF QUALIFICATION TEST METHODS FOR GEAR LUBRICANTS. D. L. Powell

April, 1961, (142) p (Contract DA-11-022-ORD-905) DA Proj. -593-21-060, OCO Project TB5-2, Unclassified Report.

A multi-phase experimental and analytical project has been undertaken to develop test procedures for the evaluation of hypoid gear lubricants and to study the loading, temperature and sliding velocity conditions which exist on automotive hypoid gears. Each of the six phases of this work are discussed in individual sections of this report.

The work included in this project conducted in the period from 1952 to 1960 is as follows:

- I. Development of Moisture Corrosion Test Technique
- II. Study of Dynamic Loading of Automotive Hypoid Gears
- III. Study of Sliding Velocity of Automotive Hypoid Gears
- IV. Study of Unit Surface Loading of Hypoid Gears
- V. Study of Dynamic Surface Temperatures of Hypoid Gears
- VI. Cooperation with the CRC Gear Lubricants Group in the Development of Test Techniques and Field and Laboratory Investigations.

UNCLASSIFIED

1. Procedure development - Gear lubricants
2. Gear Lubrication